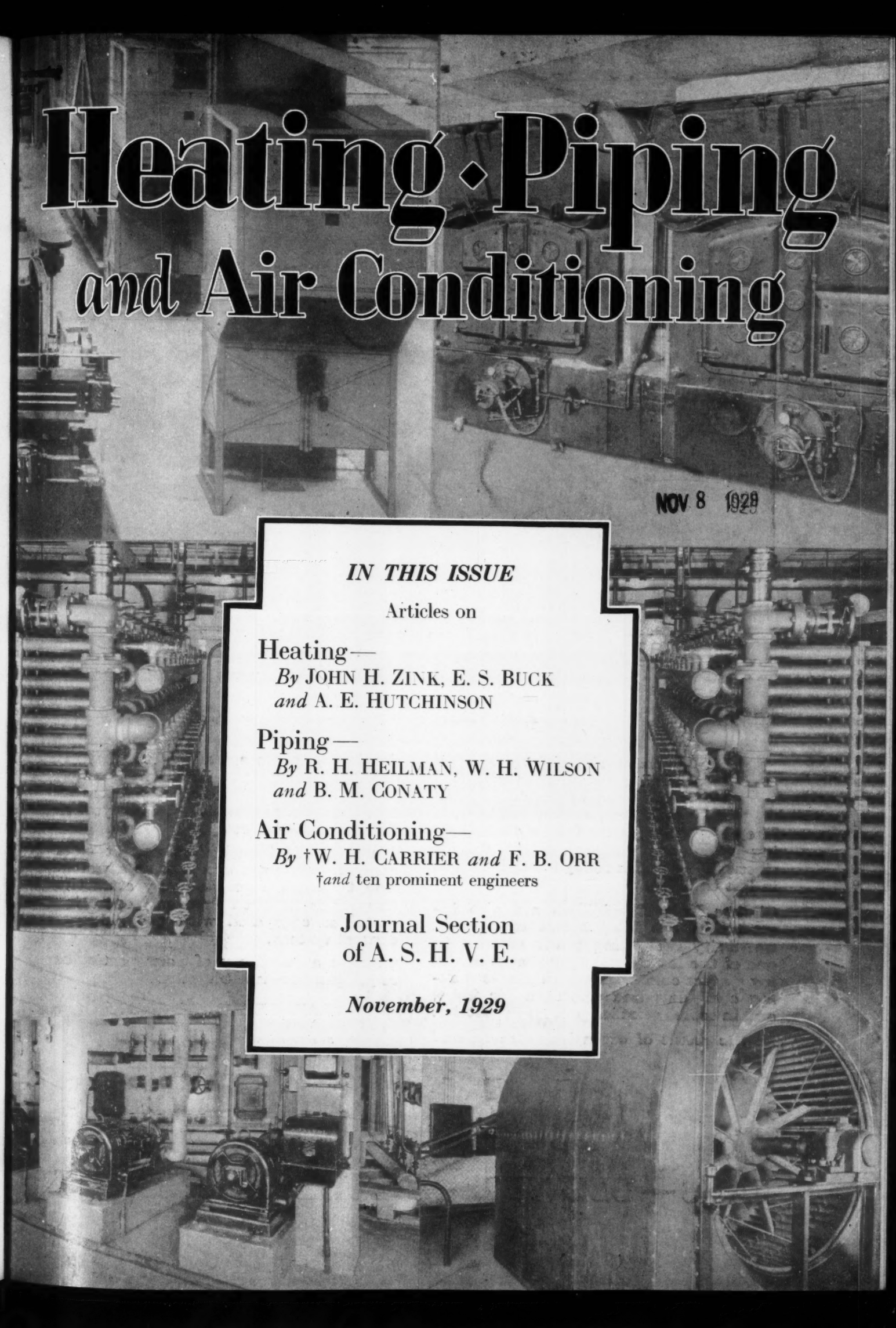


Heating · Piping and Air Conditioning

NOV 8 1929

IN THIS ISSUE

Articles on

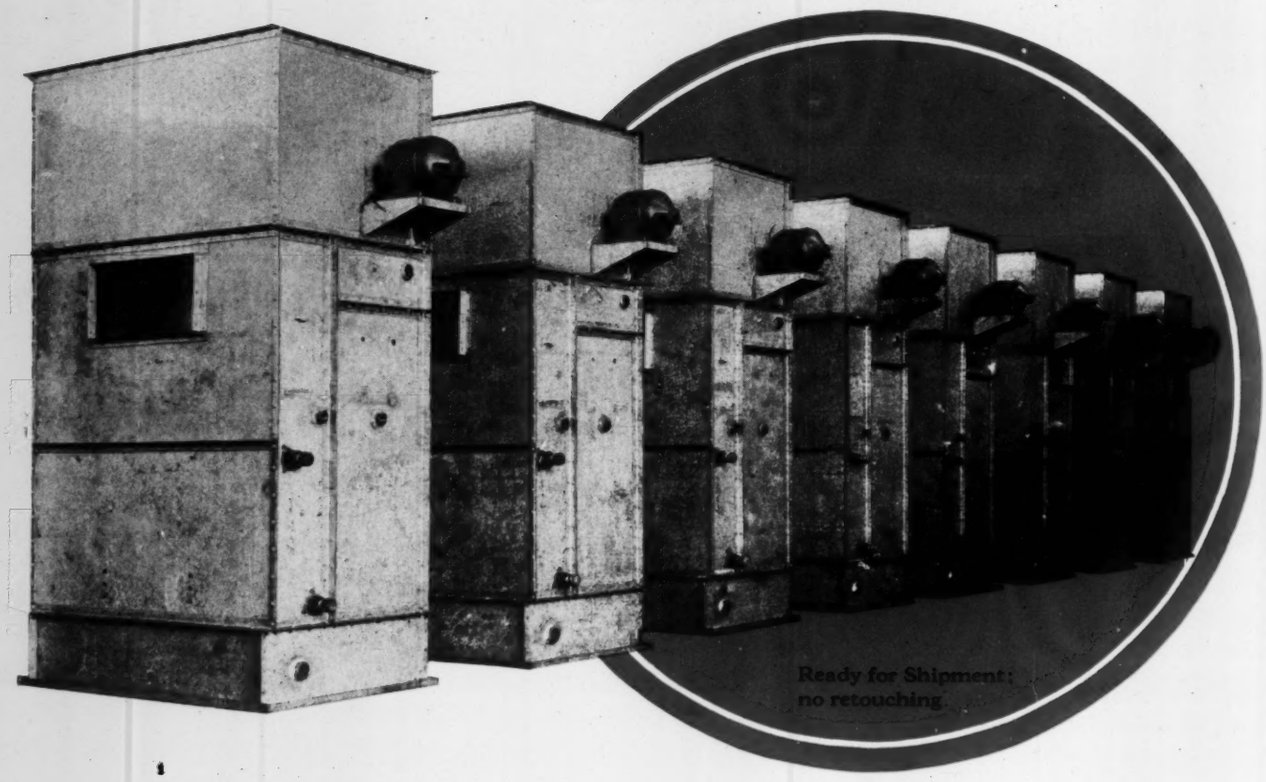
Heating—
By JOHN H. ZINK, E. S. BUCK *and* A. E. HUTCHINSON

Piping—
By R. H. HEILMAN, W. H. WILSON *and* B. M. CONATY

Air Conditioning—
By †W. H. CARRIER *and* F. B. ORR
†*and* ten prominent engineers

Journal Section
of A. S. H. V. E.

November, 1929



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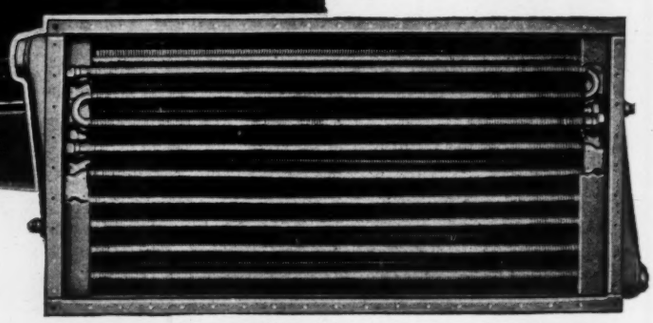

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

HOW MUCH INSULATION?

In every field of design or construction the question of initial investment versus the saving in operating costs appears. It is no different when the insulation of piping is concerned. How much may be spent for insulation in order to reduce the inevitable heat losses?

R. H. Heilman, senior fellow at the Mellon Institute of Industrial Research and no stranger to readers of these columns, is the author of an article on the "Economic Thickness of Heat Insulation" which appears in this issue. The method of calculating the proper amount of insulation to apply figured on the basis of obtaining the lowest annual operating expenses is discussed by Mr. Heilman in his paper.

Much of the valuable information contained in the article is condensed and appears in chart form. These charts should be of great interest to everyone concerned with the insulation of piping. A table which shows the capitalized cost of ten B.t.u.'s per hour for a wide variety of fuel costs will also prove of use to engineers, contractors and others.

TECHNICAL MATERIAL TO BE MADE AVAILABLE READILY

Of over six hundred pages of editorial material published to date in seven issues of HEATING, PIPING AND AIR CONDITIONING, approximately twenty-five per cent has been devoted to heating, forty per cent to piping and thirty-five per cent to air conditioning. These proportions were agreed upon by our editorial and business staffs after an exhaustive survey of our subscribers, and is designed to represent their interests.

An index to appear in the December issue will classify this wealth of material so that it will be readily available to those possessing files of this publication. If one takes the time to glance over the back numbers of HEATING, PIPING AND AIR CONDITIONING he will quickly realize that such an index will prove extremely useful on many occasions.

A surprising fact, brought forcibly to our attention while compiling a classification of the articles published up to the present time, is the number of contributions made to the literature of engineering in the short span of seven issues. Quantity has not been increased at the expense of quality, though.

CARRIER AND A. S. H. V. E. COMMITTEE REVIEW AIR CONDITIONING

"The Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort" is the complete title of an equally complete paper prepared by Willis H. Carrier and ten prominent engineers comprising an assisting committee appointed by the American Society of Heating and Ventilating Engineers.

This paper was prepared for presentation at the World Engineering Congress in Tokio; it describes the present development of air conditioning in the United States. It is our lead-off article.

Welding Standards and Practice

John H. Zink, chairman of the Heating and Piping Contractors' National Association committee on welding, is represented in the November HEATING, PIPING AND AIR CONDITIONING by an article on welding standards and practice.

The author discusses welding practice in the United States. Articles on welding invariably create much interest; we are sure that this one will too.

Units Heat Large Plant

Fifty-three units heat the large one-story plant of the Leland Electric Company in Dayton, Ohio. Each forty by forty foot area in this structure has individual temperature control.

Everett S. Buck, consulting engineer on heating and air conditioning and connected with the General Iron Works of Cincinnati, tells of the Leland Electric installation in "Direct Fired Units Heat Large Factory."

Bad Air and Health

Every day we realize more and more the injurious effects bad air has on human beings. "Interviews of Interest," a department which appeared for the first time in the September number, stresses this basic phase of air conditioning this month.

Our correspondent travels throughout the country seeking interviews from heating, piping and air conditioning experts. This department, which has already been very favorably received, should not be overlooked in this or any issue.

A Large Pipe Line

A large acetylene gas line recently installed for the Shawinigan Chemicals Company forms the subject of an article by B. M. Conaty in this month's HEATING, PIPING AND AIR CONDITIONING. Because of the low pressure in the pipe and the duty on fabricated pipe (the plant is located in Canada) the line was made of sections rolled and welded in the field.

As the losses in two smaller lines, which were replaced by the new pipe, were rather high, the installation will pay for itself in a very short time.

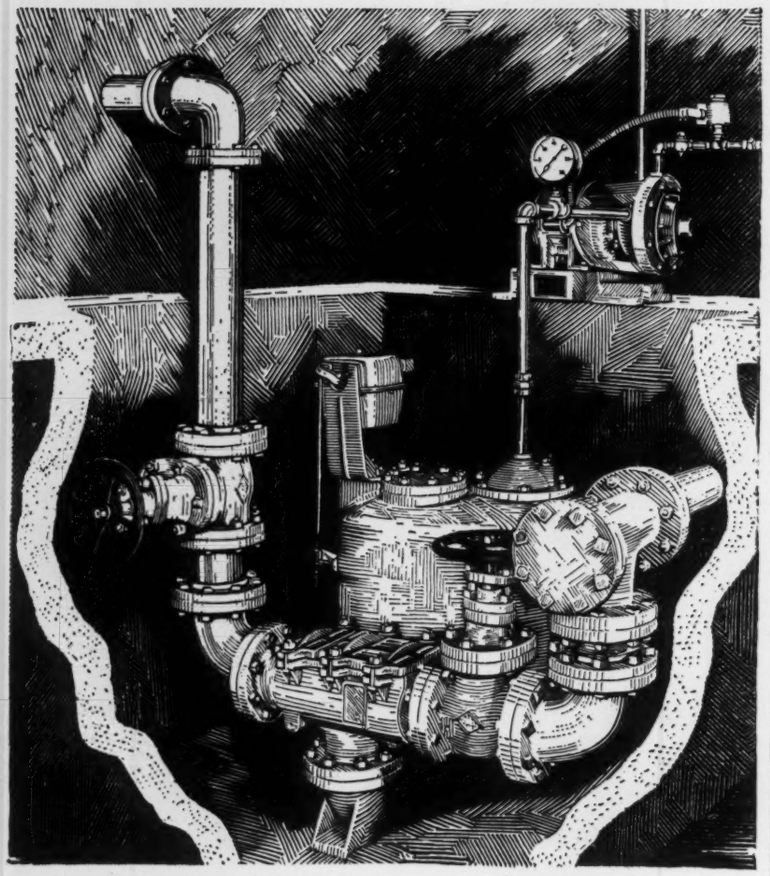
Comfort for Customers Stimulates Sales

The air conditioning of an existing building presents many problems that are not encountered in designing a similar system for a new structure. Very often there is little, or no, leeway given the air conditioning engineer or contractor to change the appearance of at least part of the building.

Such was the case when it was decided to improve the atmosphere in the Commonwealth Edison Electric Shops in Chicago; it was necessary to preserve the original architectural treatment.

Other problems, too, complicated the job. Large lamp loads, and an adjacent boiler room, were responsible for heat loads greater than those usually found. Then, too, heat distribution was uneven.

F. B. Orr, a newcomer to our pages, has written an article on this installation which will, we are certain, be read with profit by all interested in air conditions as applied to the comfort of persons and the stimulation of sales.



This small Jennings Sewage Ejector has a capacity of 30 g.p.m. Other sizes range in capacity from 50 to 1500 g.p.m. with heads up to 50 ft.

Cannot clog—

handles anything that will pass through a 4" pipe

For handling small quantities of sewage, the 30 gallon Jennings Ejector is an inexpensive unit which operates at low cost.

Similar in construction to the larger Jennings units, this small ejector works pneumatically without employing air valves, reciprocating compressors or other complicated devices. Screens are unnecessary and, since no vital working parts come in contact with the sewage, efficiency can never be impaired by parts becoming caked with solid matter. Extra large inlets and outlets, four inches in diameter, are provided, permitting the ejector to handle large solids without clogging.

This 30 gallon Jennings unit is just the ejector for installation in basements of apartment houses, hotels and public buildings where it is necessary to raise sewage and drainage from toilets and other fixtures to street sewer level automatically and with little or no attention. The receiving pot may be placed in a pit in any out-of-the-way corner, the Nash Hytor compresser and driving motor on the basement floor level, or on wall brackets, where they are always accessible.

Jennings Pumps

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Heating · Piping and Air Conditioning

Vol. I November, 1929 No. 7

The Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort†

By Willis H. Carrier[1]

This paper deals with the theory and practical application of air conditioning for the purposes of industry and to assure conditions of human comfort especially in factories and auditoriums.

The industrial importance of modern air conditioning is described and typical illustrations are given of its economic value.

Modern air conditioning has largely developed since 1905 and its expansion has been rapid.

The physical principles underlying the art of air conditioning and psychrometry are briefly reviewed and the following four fundamental principles are stated and demonstrated:

1. When dry air is saturated adiabatically the temperature is reduced as the absolute humidity is increased, and the decrease of sensible heat is exactly equal to the simultaneous increase in latent heat due to evaporation.
2. As the moisture content of air is increased adiabatically, the temperature is reduced simultaneously until the air is saturated, when no further heat metamorphosis is possible. This ultimate temperature may be termed the temperature of adiabatic saturation.
3. When an insulated body of water is permitted to evaporate freely in the air, it assumes the temperature of adiabatic saturation of that air and is unaffected by convection, i. e., the true wet bulb temperature of air is identical with its temperature of adiabatic saturation.
4. The true wet bulb temperature of the air depends entirely on the total of the sensible and the latent heat in the air and is independent of their relative proportions. In other words, the wet bulb temperature of the air is constant, providing the total heat of the air is constant.

Various relationships based upon these principles of total heat, relation of dew point to relative humidity and the determination of the moisture content from wet and dry bulb temperature readings are shown. The laws and experimental data involving the rate of evaporation of water in contact with air are given.

The experimental relationships of temperature, humidity and air motion to human comfort are discussed and the more important experimental data with reference to effective temperatures and the relation to human comfort as determined by the Research Laboratory of the A. S. H. V. E. and the Harvard School of Public Health are given.

There is also given an analysis of the heat given off by the human body in the two forms of sensible and latent heat under various conditions of temperature and humidity. This is important in determining the requirements of air conditioning in many types of equipment.

The various types of equipment used in modern air conditioning are described. These comprise briefly humidifiers, dehumidifiers, and instruments responsive to changes in temperature or humidity employed in automatic control.

Systems of air distribution are also described and the importance of engineering in this field is emphasized.

THE control of atmospheric conditions within an enclosure with reference to temperature, humidity, and cleanliness is termed *air conditioning*.

The importance of this field of engineering has recently become widely recognized, not only in engineering circles, but by the public at large. Perhaps this is best evidenced by the fact that the forthcoming edition of the Encyclopedia Britannica will contain a chapter devoted exclusively to air conditioning. Such chapters are already to be found in some of the leading engineering hand-books.

It has long been recognized that relative humidity is

† Presented at the World Engineering Congress, Tokio.

[1] Assisted by a Committee appointed by the American Society of Heating & Ventilating Engineers as follows:

Prof. A. C. Willard, W. L. Fleisher, S. R. Lewis, F. R. Still, O. W. Armspach, H. P. Gant, C. P. Yaglou, H. W. Ellis, F. C. Houghten, L. A. Harding

an important factor in the manufacture and processing of certain hygroscopic materials such as textiles. Since the normal relative humidity in textile factories is nearly always lower than that desired, various artificial means have for many years been provided to increase the humidity.

On the other hand, there are industries which require a definite and unvaried humidity so that at some periods the normal quantity of moisture in the air must be increased, and at other times lowered. Among such industries may be mentioned the manufacture of confectionery, the processing and weaving of artificial silk, and the printing and lithographing industry.

Other industries require not only a constant relative humidity, but also a fairly uniform temperature. An example of this is seen in the modern automatic wrapping machines, used for wrapping chewing gum, food products, confectionery, and machine made cigarettes, which require exact conditions of heat and moisture in order to function satisfactorily without frequent adjustments.

In the manufacturing and processing of most hygroscopic materials there are usually one or more stages in the process in which moisture has to be removed from the material. In all such products which are not in themselves soluble in water, this is usually accomplished by air drying.

It has long been recognized that relative humidity is an important factor in the manufacture and processing of certain hygroscopic materials such as textiles. Since the normal relative humidity in textile factories is nearly always lower than that desired, various artificial means have for many years been provided to increase the humidity.

In order to avoid injury to the products which require temperature and humidity control, the rate of moisture removal must be controlled with accuracy in certain stages of the process, while at the end of the process of drying it may be extremely important to control with accuracy the final moisture content of the material. This is true in the manufacture of the various tobacco products, artificial silk, certain gelatine products including photographic films, lumber, paper, and macaroni. In many of these products it is not only necessary to have this final moisture content to insure the desirable qualities of the product itself, but to standardize its moisture content from the standpoint of sales value, as for example, in artificial silk of which hundreds of millions of dollars worth is manufactured annually, 1 per cent variation in moisture content would mean millions of dollars in the variation of the sale price of the product, although not in its intrinsic value. The same is true to a lesser extent in cotton yarns with which an increased moisture content is not only desirable from the sales standpoint but from the manufacturing standpoint as well. Wheat flour is another product that requires careful moisture standardization and considerable attention is being given to this field at the present time.

Air conditioning is important in certain branches of the chemical industry in controlling the temperature of reaction and facilitating or retarding evaporation.

The control of the moisture content of air supplied to blast furnaces in the manufacture of pig iron has proven advantageous, and with present improved methods of moisture removal from the air, this field may become prominent.

In many instances, air conditioning combined with ventilation may be commercially desirable from the standpoint of human comfort, rather than primarily for the effect upon the products themselves. In many industries the output is affected, and in some cases limited, by the capacity of the workers. This output is therefore greatly augmented by the proper control of temperature and humidity conditions, when normally they would be so unsatisfactory as to depress in marked degree the factory output. This is particularly true in the cotton mill industry, in steel rolling mills, in hot climates like India, in deep mines particularly in Brazil, India, and South Africa, in the manufacture of glazed iron ware and similar processes.

For instance in such industries as lithographing, the question of the quality of the product and the personal comfort of the employees assume almost equal importance. So that the proper register may be obtained, with the application of the colors in different processes, a very exact humidity is necessary, and in hot moist weather such conditions can only be secured by dehumidification and artificial refrigeration, except at the price of great human discomfort. In other industries, the product is affected by both relative humidity and temperature, as in the manufacture of chocolates, chewing gum, cigarettes and machine made cigars, where it is necessary to humidify during some parts of the year, and to dehumidify and refrigerate at other seasons, in order to maintain a uniform condition both of temperature and humidity within the limits permissible by the products. It happens in these instances, as in many others, that the atmospheric conditions best suited to the products and process are almost ideal for the health, comfort and efficiency of the workers.

Besides the industrial uses of air conditioning, it has great commercial importance in its application to personal comfort alone, where large numbers of people are congregated as in large auditoriums, such as cinema theatres and in large department stores. In such cases, ventilation with cooling and dehumidification have been proven to offer high economic returns.

In one of the early cinema installations, the cost of installing such equipment was approximately $65,000.00, and the increased box office receipts during the first summer's operation over the previous summer, without such a system, exceeded $70,000.00. Hundreds of installations have been made throughout the United States and others will be found in London, Paris, and Havana. The use of such installations will, undoubtedly, rapidly increase in other parts of the world.

Air conditioning with refrigeration has been found practically the only satisfactory method of securing the desired degree of comfort during days of special summer sales and at other rush seasons in large department

stores. It has been found to add materially to the profits in the operation of such stores. One of the large stores in Detroit was the first to install a modern air conditioning system with refrigeration. In this initial installation the refrigeration capacity was 600 tons. A large addition to the same store required over 1,000 tons of refrigeration in conjunction with the air conditioning system for the same purpose, and when it is remembered that refrigeration costs nearly $200.00 a ton to install and that the cost of the refrigerating equipment represents approximately one-third of the entire installation, the dollars and cents value to this firm of such an installation can be appreciated.

The use of air conditioning with refrigeration in hotels, restaurants, banks, and office buildings is likewise rapidly increasing.

Air conditioning as applied in textile mills has made possible, in the United States, a shifting of the center of this industry from the New England States in the North where the climatic conditions are advantageous to the less favorable climatic conditions of the Southern States, but where the cotton is grown and labor is available.

By means of air conditioning the desirable conditions of temperature and moisture are maintained in one climate equally as well as in another. In tobacco factories it has resulted in a saving which, in some instances, paid for the equipment required in less than one month's operation due to the elimination of waste material and general improvement in efficiency. A great speeding-up of operation in large cigarette factories throughout the United States and avoiding shutdowns and delays followed the use of air conditioning, while in textile factories it has permitted continuous operation without the delays required in the adjustment of loom fixtures for variable atmospheric conditions. In such industries, the accurate control of humidity at a definite point is of as much, or more, importance than the humidity itself.

On the other hand, there are industries which require a definite and unvaried humidity so that at some periods the normal quantity of moisture in the air must be increased, and at other times lowered. Among such industries may be mentioned the manufacture of confectionery, the processing and weaving of artificial silk, and the printing and lithographing industry.

History of the Development of the Air Conditioning Art

Prior to 1905 there had been numerous devices used, principally in textile mills, for atomizing water into the air to increase the moisture content. These were controlled by turning on and off such devices manually. A few spray air-washers were in use in public buildings which likewise added to the moisture content of the air. There was, however, an entire lack of definite methods of humidity control, nor were these devices applied industrially.

Subsequent to this date, Johnson, Cramer & Hodge, Carrier, Comfort, and others, devised various practical methods and instruments for the definite control of relative humidity. The air-washer was improved to such an extent that it produced complete saturation of the air and was applied in the ventilation, cooling and humidification of textile mills and other industrial establishments.

The spray type of dehumidifier also made its appearance about this time. In this system the air was cooled and the moisture extracted therefrom by a specially designed type of air washer in which the effect was produced by a direct spray of artificially cooled water upon the air rather than through coils of pipe in which cold water or brine was circulated, as had been the method theretofore. All the water vapor was thus condensed from the air to the point of saturation at the temperature corresponding to that of the air leaving the dehumidifier and all free moisture was removed by a series of baffles or eliminators, in much the same manner as in the standard air-washer. The spraying of water into the air to remove moisture from it might seem paradoxical except for the application thereto of easily understood laws of physics, of which more will be said later. This type of dehumidifier came gradually into use and is now applied to a large number of industries, and to air conditioning for human comfort in theatres, department stores, and similar places.

Today the use of refrigerated surfaces such as pipes for the purpose of cooling and dehumidifying air have given way almost entirely to the use of the direct spray of cooled water. One of the principal reasons for this change is the high effectiveness of heat transfer between the finely divided spray and the air which is thereby lowered substantially to the temperature of the leaving water. This greatly reduces the temperature head required in refrigeration and, therefore, the size and cost of the refrigerating equipment. This process was used prior to 1923 in the cooling of a number of theatres but without any attempt to control the humidity.

It was not until 1923 that the problem of cooling theatres economically and, at the same time, controlling the humidity independent of the cooling was solved. This process is in general use today in all auditorium applications or where human comfort is involved.

The problems involved in auditorium cooling, and the method of accomplishing the results, are quite different from the method applied to manufacturing processes.

Physical Principles

The art of air conditioning is based upon very definite physical principles. The phenomena involved have been given intensive study, both from the theoretical and experimental standpoint. As the result of this, the various relationships have been quantitatively expressed, either rationally or empirically.

The usual requirement in air conditioning as applied to processing hygroscopic materials is to maintain a definite relative humidity. In other instances it is desired to control merely a definite moisture content of the air.

In problems involving human comfort, it is necessary

to maintain certain limiting or desirable effective temperatures which depend upon the experimentally determined relationships of wet and dry bulb temperatures and air motion. Briefly, those are the requirements in various phases of air conditioning and the terms should properly be defined in order that the following discussion may be clear.

It should be noted first that the moisture content in a given space is independent of the maximum quantity of air or other gases present. In other words, the properties of saturated water vapor are assumed to be in no way affected by the presence of other gases. Therefore the quantity of moisture present in a given space may be determined directly from the data available or the physical properties of water vapor.

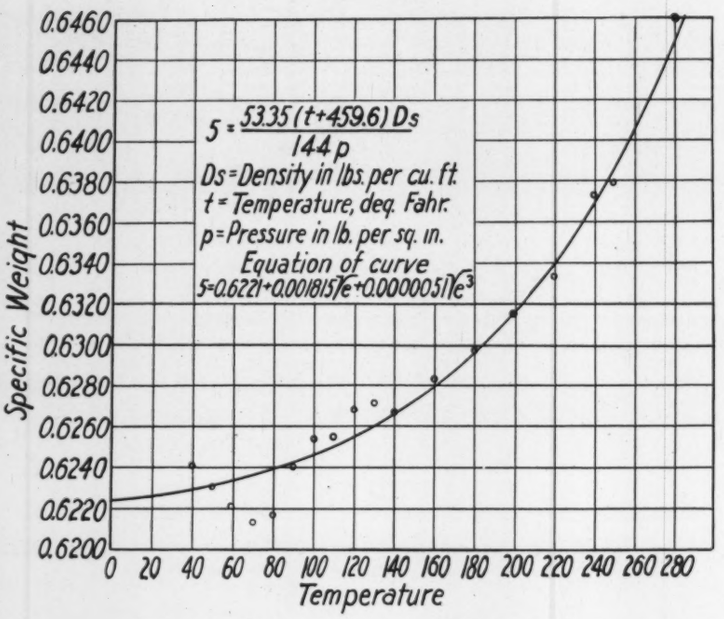


FIG. 1. SPECIFIC WEIGHT OF WATER VAPOR

As is well known, the vapor pressure of water, and consequently the weight per unit of volume of that vapor, depends upon its temperature. Vapor pressure and density curves and tables are available for all the ranges of atmospheric temperature, and the accuracy of these can probably be relied upon.

Where the vapor pressure of the water vapor present is known, it is evident that the proportionate weight of water vapor to other gases in the mixture obeys Dalton's law assuming, of course, that they have no chemical affinity. This may be expressed by the relationship:

$$W = \frac{Se}{P - e}$$

where

W = weight of moisture per pound of air.

S = specific weight of water vapor compared with air as 1.

P = barometric pressure.

e = vapor pressure at temperature t.

If other gases than air are present, then the specific weight of water vapor must be taken as the density of the water vapor with reference to the density of the gas at the same pressure and temperature.

The theoretical specific weight of steam, assuming that it obeys the gas laws, would be the ratio of the molecular weight of steam to that of pure dry air. At low temperatures the actual specific weight approaches very closely this theoretical, while at high temperatures the deviation is considerable. The curve of actual specific weights of water vapor is given in Fig. 1.[2]

From this, it is possible to compute with accuracy the weight of water vapor contained in a pound of dry air at any given vapor pressure and barometric pressure.

[2] Rational Psychrometric Formulae, *Trans. A. S. M. E.*, 1911, Vol. 33, p. 1020.

In a mixture of air and water vapor, a point is reached where there is a maximum quantity of water vapor present for any temperature; that is, the air is saturated when the vapor pressure corresponds to the temperature.

Air, however, is usually deficient in water vapor. That is, it is unsaturated, in which case the vapor pressure is lower than the maximum corresponding to the temperature of the mixture. The ratio of such vapor pressure to the vapor pressure of the mixtures when saturated at that temperature is termed the *relative humidity*. That is:

$$\text{Per cent relative humidity} = \frac{e}{e_t} = \frac{D}{D_t} \text{ (approximately)}$$

where e and D are the pressure and the density respectively of the vapor in the air, and e_t and D_t are the saturation pressure and density respectively of the vapor corresponding to the temperature t of the mixture.

Dew Point

If such an unsaturated mixture of air and water vapor be cooled, without the removal of any moisture present, it will ultimately become saturated. This temperature is termed the *dew point*, for any further cooling will result in a deposition of moisture from the mixture.

Temperature of Evaporation

In the instance cited, the air was saturated by withdrawal of heat and without change in the moisture content.

Air may also be saturated adiabatically, that is, by being brought into contact with the liquid itself and without any addition or subtraction of external energy. Under this condition water vaporizes into the space occupied by the air, and assuming that no heat is available from the water itself. That is, no temperature change takes place in the water present, the heat required for change of state must come from the sensible heat in the air itself, thus resulting in the lowering of its temperature.

The moisture content of the air is, therefore, increased and its temperature lowered simultaneously, and it is obvious that the increase in latent heat must exactly equal the decrease in the sensible heat of the mixture. Further, this process will go on until the air becomes saturated where no further exchange between the sensible and latent heat is possible. This final temperature

at which adiabatic saturation takes place in any body of air of known temperature and moisture content is known as the *temperature of evaporation* and corresponds to the *wet bulb temperature.*

So that no heat exchange may take place between the water and the surrounding air, it is necessary that the water be introduced at this temperature of evaporation. On the other hand, if free water is exposed to a continuous current of air of such a given temperature and moisture content, it will ultimately assume of its own accord this equilibrium temperature of evaporation. This is not only logical, but has been proved experimentally. It is for this reason that an ordinary thermometer which has its bulb covered by a wetted cloth, or otherwise coated with the liquid will, if protected against outside sources of heat such as radiation and subjected to a stream of air having a definite temperature and moisture content, indicate the temperature of adiabatic saturation. This proves the identity between what is termed the *wet bulb temperature* and the *temperature of adiabatic saturation.*

In the ordinary sling psychrometer, which is used in measuring the moisture content of air, however, there is always some radiation present and, if the wet bulb be not strongly ventilated, this error may be considerable. With strong ventilation, however, the radiation effect is usually less than 1 per cent. This error depends upon the temperature and velocity of air movement, and the relationship of this error to these factors is shown in a paper presented before the *American Society of Mechanical Engineers* in 1924.

From the foregoing discussion, four fundamental principles in air conditioning may be evolved.

1. When dry air is saturated adiabatically the temperature is reduced as the absolute humidity is increased, and the decrease of sensible heat is exactly equal to the simultaneous increase in latent heat due to evaporation.
2. As the moisture content of air is increased adiabatically, the temperature is reduced simultaneously until the air is saturated, when no further heat metamorphosis is possible. This ultimate temperature may be termed the temperature of adiabatic saturation.
3. When an insulated body of water is permitted to evaporate freely in the air, it assumes the temperature of adiabatic saturation of that air and is unaffected by convection, i. e., the true wet bulb temperature of air is identical with its temperature of adiabatic saturation.
4. The true wet bulb temperature of the air depends entirely on the total of the sensible and the latent heat in the air and is independent of their relative proportions. In other words, the wet bulb temperature of the air is constant, providing the total heat of the air is constant.

It will be obvious in considering the foregoing process that a formula relating the latent heat change to the sensible heat change establishes the relationship of this theoretical wet bulb temperature to the temperature and moisture of the air. This was first pointed out in the rational psychrometric formula:[3]

$$r'(W'-W) = C_{pa}(t-t') + C_{ps}W(t-t')$$

in which

$(t-t')$ = the true wet bulb depression

$(W'-W)$ = moisture absorbed per pound of pure air when it is adiabatically saturated from an initial dry bulb temperature t_o and an initial moisture content W

C_{pa} = mean specific heat of air at constant pressure between temperature t and t'

[3] Rational Psychrometric Formula, *Trans. A. S. M E.*, Vol. 33, 1911.

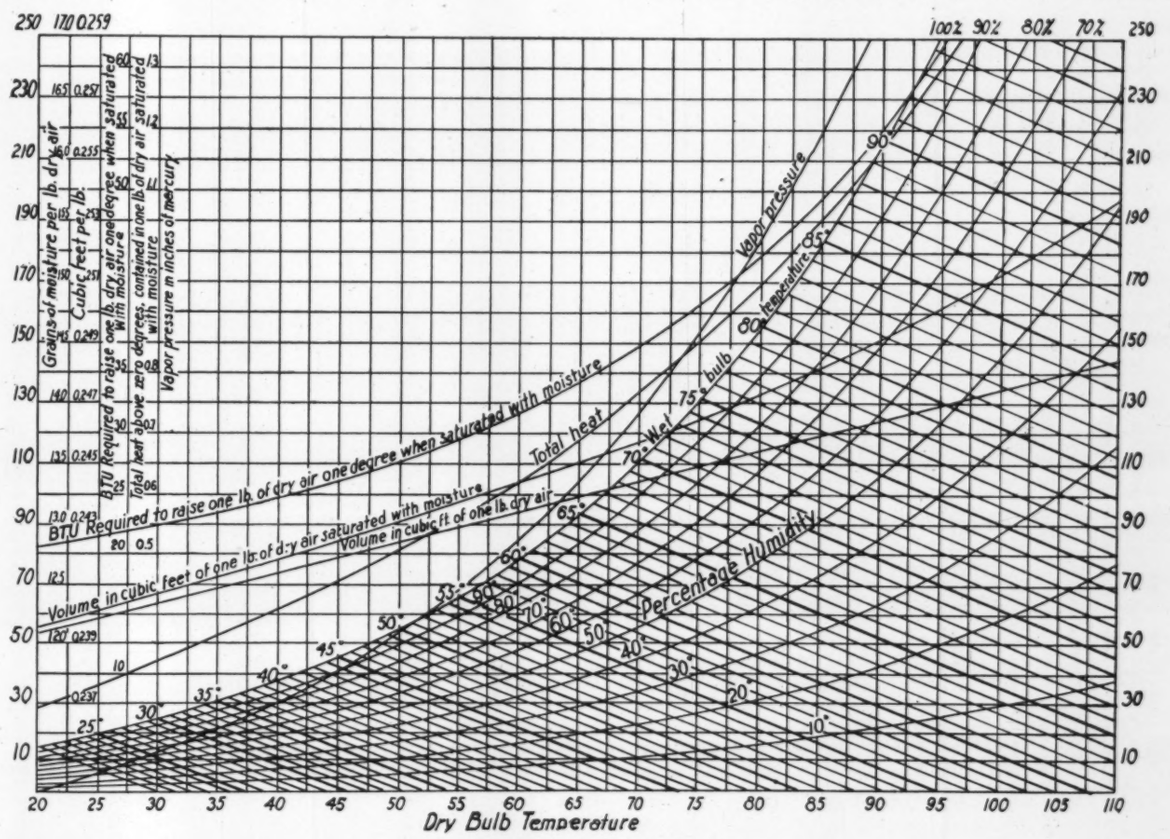

Grains of moisture per lb. dry air
Cubic feet per lb.
B.T.U. Required to raise one lb. dry air one degree when saturated with moisture
Total heat above zero degrees, contained in one lb. of dry air saturated with moisture
Vapor pressure in inches of mercury
B.T.U. Required to raise one lb. of dry air one degree when saturated with moisture
Total heat
Volume in cubic ft. of one lb. dry air
Volume in cubic feet of one lb. of dry air saturated with moisture
Vapor pressure
Wet bulb temperature
Percentage Humidity
Dry Bulb Temperature

FIG. 2. PSYCHROMETRIC CHART

C_{ps} = specific heat of steam at constant pressure between t and t'

r' = latent heat of evaporation at wet bulb temperature t'

Knowing any two of the three important values of t, t' or W, it is easy to solve for the third or for any other required relation.

This is an exceedingly useful relationship, not only in air conditioning and the drying of materials but in the psychrometry of air or other gases to which it directly applies. It is evident that if the wet bulb temperature of air of a known dry bulb temperature but unknown moisture content be determined experimentally with the wet bulb thermometer, it is possible to calculate directly the moisture content of that air from the relationship cited.

From this relationship, a psychrometric chart in common use (Fig. 2) has been evolved. This is most useful in all problems pertaining to air conditioning, drying, and in the determination of the moisture content of air.

A formula has also been derived which will permit the approximate calculation of the vapor pressure from the wet bulb temperature observation. This is useful particularly in determining relative humidities for conditions other than standard barometric conditions for which the chart is made. The formula for expressing this relationship is:

$$e = e' - (P\text{-}e')\ (t\text{-}t')\ 2800\text{-}1.3t'$$

where

e = partial pressure of the moisture in the air, which also equals vapor pressure corresponding to the dew-point.

e' = the vapor pressure corresponding to saturation at wet bulb temperature t'

P = the barometric pressure

t = dry bulb temperature in deg. fahr.

t' = wet bulb temperature in deg. fahr.

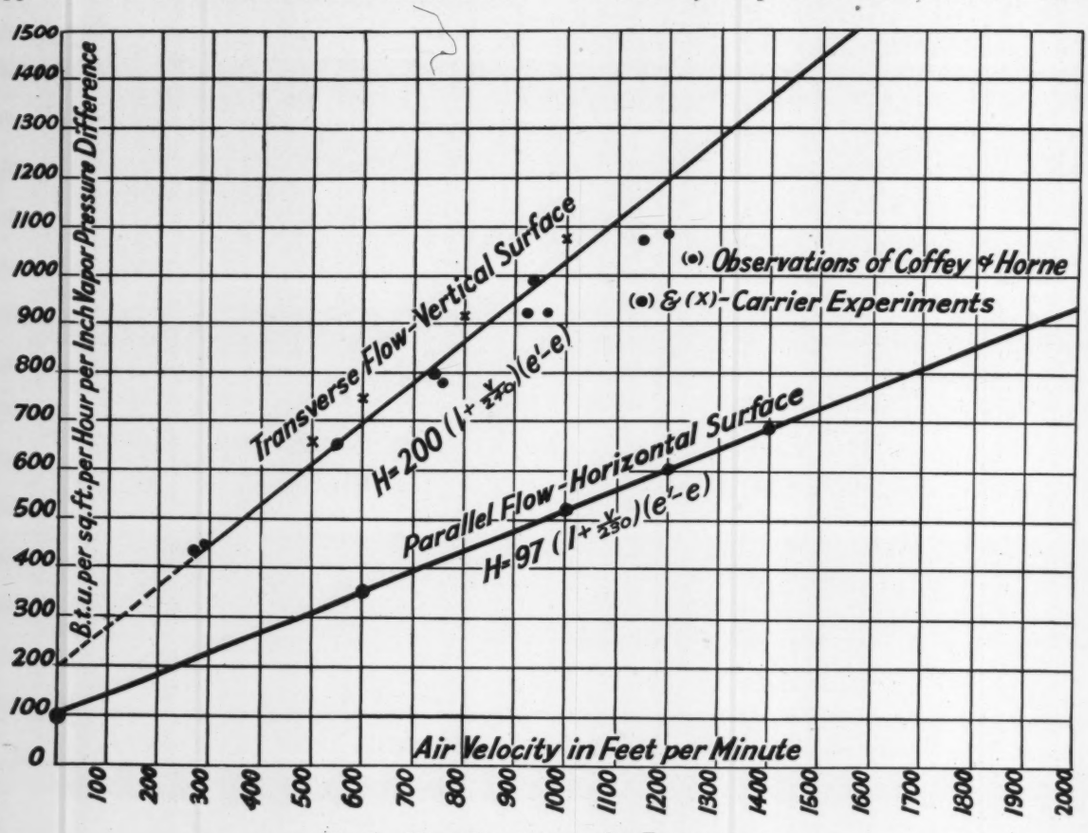


FIG. 3. HEAT TRANSMITTED BY EVAPORATION

Total Heat

In cooling and dehumidifying air, it is necessary to take into consideration not only the sensible heat of the air itself but the latent heat of the moisture removed.

Air at any temperature and moisture content may be considered as having a certain definite total heat in thermal units per unit weight as calculated from some arbitrary base such as zero moisture and zero deg. fahr. temperature. By using the psychrometric chart, on which are given wet bulb temperatures and corresponding total heats, it is possible to determine the total heat to be removed in the process of dehumidification by subtracting the total heat of the cooled air with its reduced moisture content from that of the air in its initial condition as indicated on the chart. This process is greatly simplified by taking into account the fourth psychrometric principle; in other words, in observing the fact that the wet bulb temperature of the air may be used as the measure of the total heat which the air contains. Thus, by referring to the chart, having given air of known temperature and known moisture content, a corresponding wet bulb temperature is obtained. Then, under standard barometric conditions, the quantity desired is determined by reference to the curve of total heat. For example, air at 85½ deg. with 80 per cent relative humidity has exactly the same total heat as air at 101 deg. and 40 per

cent relative humidity, because in both cases the wet bulb temperature is 80 deg. This wet bulb temperature represents about the maximum wet bulb temperature ever encountered in the temperate zones under extreme conditions, and is nearly the maximum found in the tropics. A normal high wet bulb temperature for the temperate zone may be taken at 75 deg. in calculating requirements for air cooling, as this is seldom exceeded except for short periods.

Relation of Dew Point to Relative Humidity

A peculiar relationship between the dew-point and the relative humidity is found most useful in air conditioning work, namely that for a fixed relative humidity there is substantially a constant difference between the dew-point and the dry bulb temperature over a considerable temperature range. The following table giving the room temperature, dew-point temperature, and dew-point differentials for 50 per cent relative humidity illustrates this relationship most clearly:

Dry bulb temperature (deg.)	65	70	75	80	85	90
Dew point temperature	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew point and dry bulb temperature	19.2	19.5	19.75	20.25	20.75	21.25

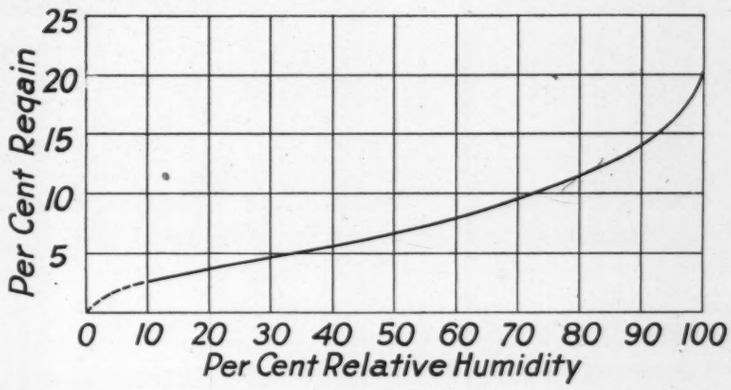


FIG. 4. REGAIN CHART FOR COTTON (SEA ISLAND) AND PAPER (CELLULOSE) AT 70 DEGREES

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg. throughout this range of dry bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg. through this range.

The approximate relative humidity for any difference between dew-point and dry bulb temperature may be expressed as:

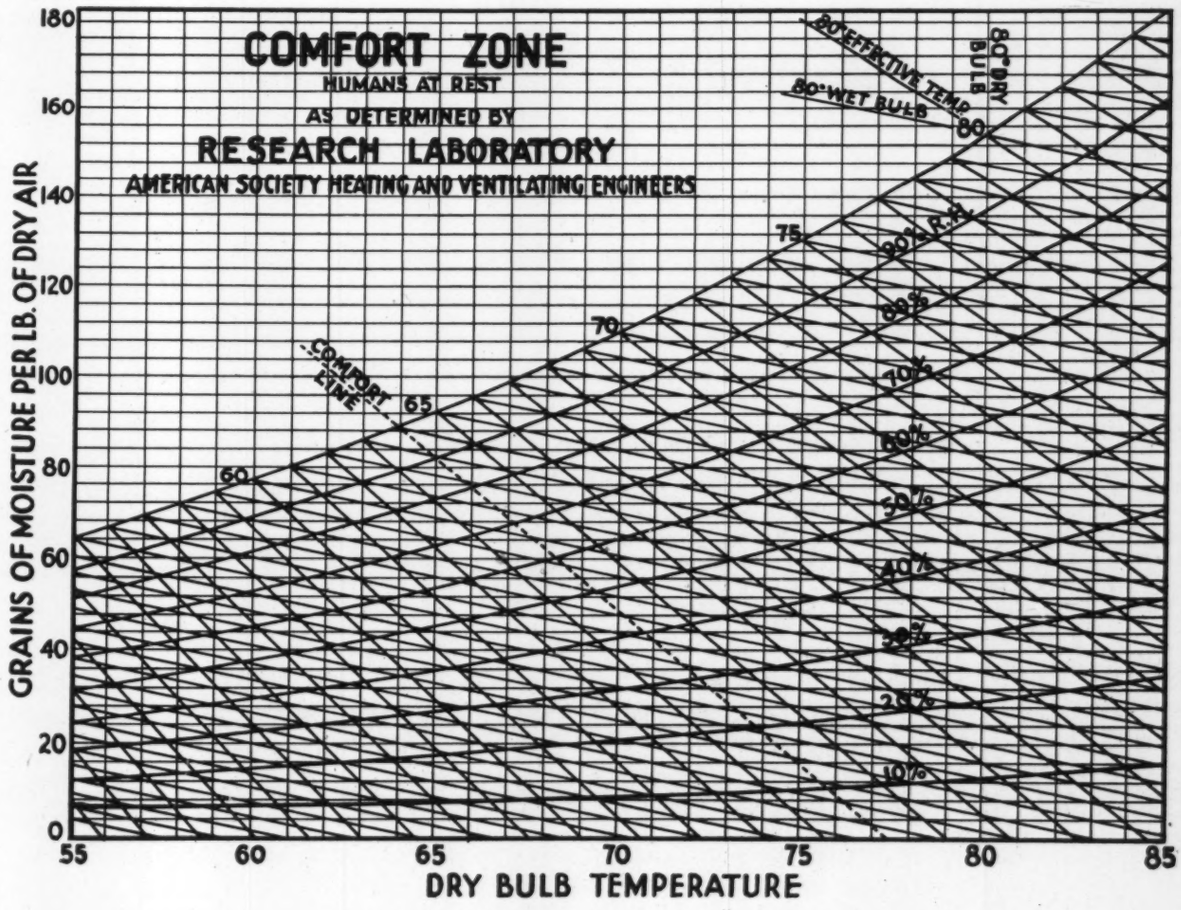


FIG. 5. THE COMFORT CHART FOR HUMANS AT REST

$$\frac{100}{\frac{t-t_1}{20}}$$
$$2$$

where t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable by the use of ventilation with saturated air when a desired relative humidity is to be maintained in a room even though there may be a wide variation in room temperature. This problem is one that applies to certain industrial applications, such as those in cotton mills, tobacco factories, etc., where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.[4]

Rate of Evaporation

In problems of air conditioning and drying, as well as in other industrial applications of evaporation, such as cooling towers, for example, it is desirable to determine the rate of evaporation. There are two distinct cases of evaporation. The *first* case is that in which the source of heat is primarily from the water itself and the air temperature may even be raised. The *second* is that in which the heat for evaporation is obtained entirely from the air itself, in which case the air is cooled and the temperature of the water remains substantially constant at the wet bulb temperature. Both cases, however, may be reduced to a common basis of calculation. It has been found that the increase in the rate of evaporation is nearly in direct proportion to the increase in air velocity, and that it is in direct proportion to the difference in vapor pressure between the vapor pressure of the water and the pressure of the vapor in the air.

The general formula[5] covering the experimental data may be expressed as follows:

$$\frac{dw}{dt} = (a + bv)\ (e' - e)$$

where

$\frac{dw}{dt}$ = rate of evaporation

a = the rate of evaporation in still air

b = the rate of increase with velocity

e' = the vapor pressure of the liquid

e = the vapor pressure in the atmosphere

v = velocity

The only difference between case *one* and case *two* is that in case *one* the vapor pressure of the liquid is one of the known or assumed factors, being dependent upon the known temperature of the liquid, while in case *two* e' is the vapor pressure corresponding to the wet bulb temperature of the air.

This wet bulb or evaporation temperature is dependent upon the dry bulb temperature and the moisture content, or upon the total heat of the air as indicated in the previous paragraph.

The effect of air velocity depends upon whether the flow of air is parallel to the surface or perpendicular to the surface elements. For a flow of air parallel to a horizontal surface

$$w = 0.093\ \left(1 - \frac{v}{230}\right)\ (e' - e)$$

(approximate) where

w = pounds evaporated per square foot per hour

v = velocity of atmosphere over surfaces in feet per minute

e' = vapor pressure of the water corresponding to its temperature

e = vapor pressure in the surrounding atmosphere

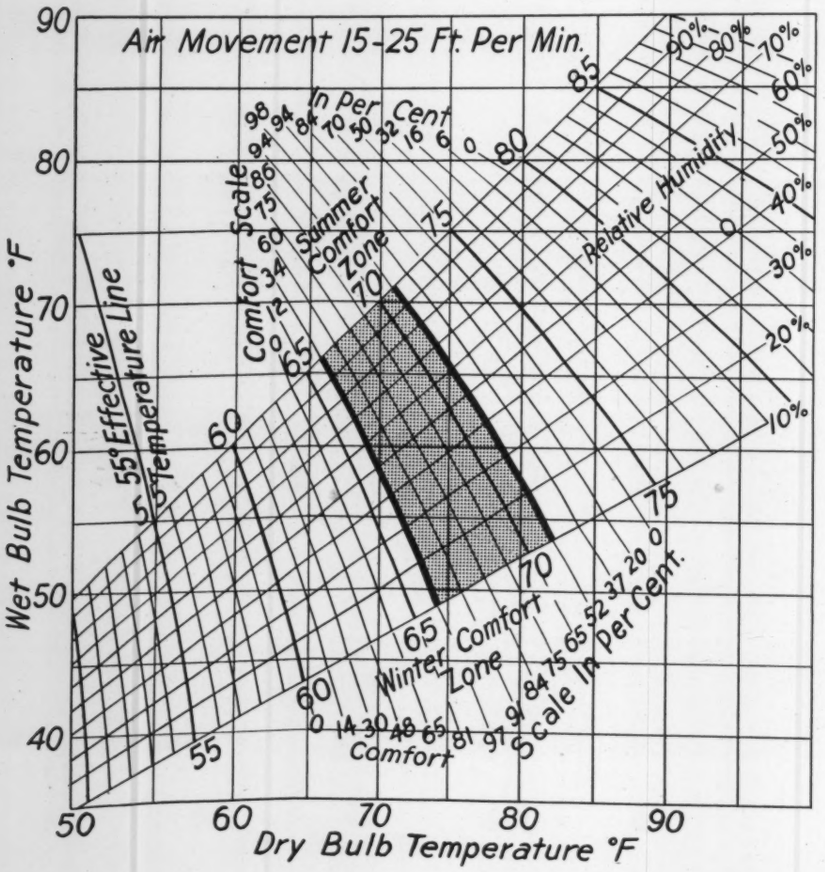


FIG. 6. EFFECTIVE TEMPERATURE CHART WITH COMFORT ZONES SUPERIMPOSED

For transverse flow, as across a tubular surface, the rate of evaporation is nearly doubled.

These relationships are indicated graphically on the chart, Fig. 3.[6]

[4] Air Conditioning Apparatus, W. H. Carrier and F. L. Busey, *A. S. M. E., Transactions* 1911, Vol. 33.

[5] The Theory of Atmospheric Evaporation—With Special Reference to Compartment Dryers. W. H. Carrier, *Journal of Industrial and Engineering Chemistry*. Vol. 13, No. 5, p. 432, May, 1921.

[6] Coffey and Horne, *A. S. R. E.* 2(1916), 5. W. H. Carrier, TRANSACTIONS A. S. H. V. E., 24(1918) 25.

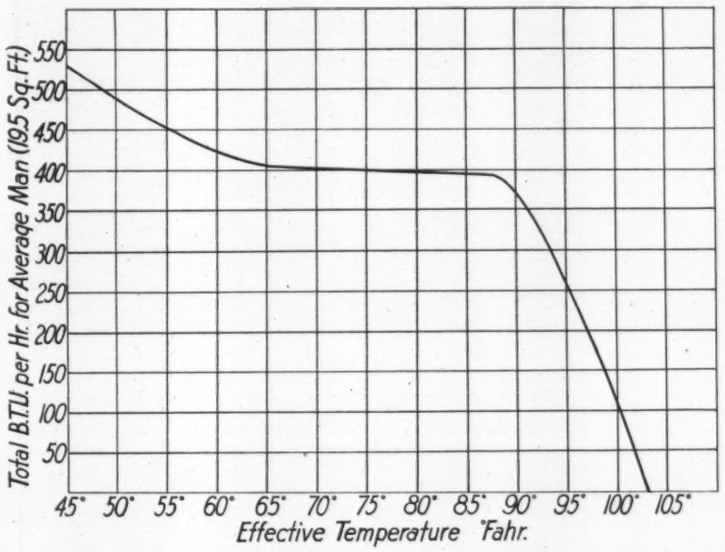


FIG. 7A. HEAT PER HOUR FOR AVERAGE MAN IN RELATION TO EFFECTIVE TEMPERATURE

Since the difference in vapor pressures is substantially proportional to the difference between the wet and dry bulb temperatures (i. e., the wet bulb depression) the rate of evaporation is also for case *two* substantially proportionate to the wet bulb depression.

In case *two*, the rate of sensible heat transfer from the air to the liquid to produce evaporation is substantially the same as the rate of heat transfer with the same type of surface without moisture being present, but with the same temperature differences. In other words, the rate of heat transfer depends upon the temperature difference only, whether the surface is wet or not. For example, it has been shown that the rate of heat transfer with air flowing across staggered coils may be represented by the formula:[7]

$$U = \frac{1}{0.0447 + \frac{50.66}{v}}$$

where

U = B.t.u. per hour per square foot per degree difference in temperature as experimentally determined from tests on heater coils.

At 400 velocity, $U = 5.8$, and at 800 velocity $U = 9.3$.

Referring to the chart showing the B.t.u. transfer in evaporation for different air velocities and transverse flow, it will be noted that there are 560 B.t.u. per hour transferred by evaporation per inch difference of vapor pressure at 400 velocity, and 910 B.t.u. per square foot per hour per inch difference in vapor pressure.

[7] Air Conditioning Apparatus. W. H. Carrier & F. L. Busey, *Trans. A. S. M. E.* Vol. 33, 1911.

[8] The Theory of Atmospheric Evaporation, With Special Reference to Compartment Dryers. W. H. Carrier, *Journal of Industrial and Engineering Chemistry*, Vol. 13, No. 5, p. 432, May 1921.

It has been pointed out[8] that 1 in. of vapor pressure difference corresponds approximately to 95 deg. difference between the wet and dry bulb temperature. Dividing by 95 the value of 5.9 B.t.u. per square foot per degree difference in temperature is obtained for 400 velocity and 9.55 B.t.u. per square foot.

It will be noted that for these two cases the heat transfer by evaporation per degree difference in temperature corresponds almost exactly to the heat transfer by convection coils. The similarity may be noted by comparing the formula[9] for heat transfer in parallel flow where

$$U = \frac{1}{0.026 + \frac{161}{v}}$$

with the heat transfer in evaporation with parallel flow. The relationship will be seen to be very close in both cases and would indicate that the heat transfer in evaporation is actually brought about by a process of convection.

The difference of the formulas may be due to errors in observation at the higher and lower velocities.

In cooling air and condensing out the moisture therefrom the heat transfer is considerably more rapid than when the air is dry and no moisture is condensed. In general, the rate of heat transmission on the air side is increased an amount which is proportionate to the latent heat removed as compared with the sensible heat removed. That is, if the latent heat removed was 50 per cent of the sensible heat removed, then the conductivity of the surface in contact with the air would be increased approximately 50 per cent.

Hygroscopic Properties of Materials

Hygroscopic materials are those which readily absorb or give off moisture to the surrounding air.

(*Continued on Page* 605)

[9] *Fan Engineering*, Second Edition.

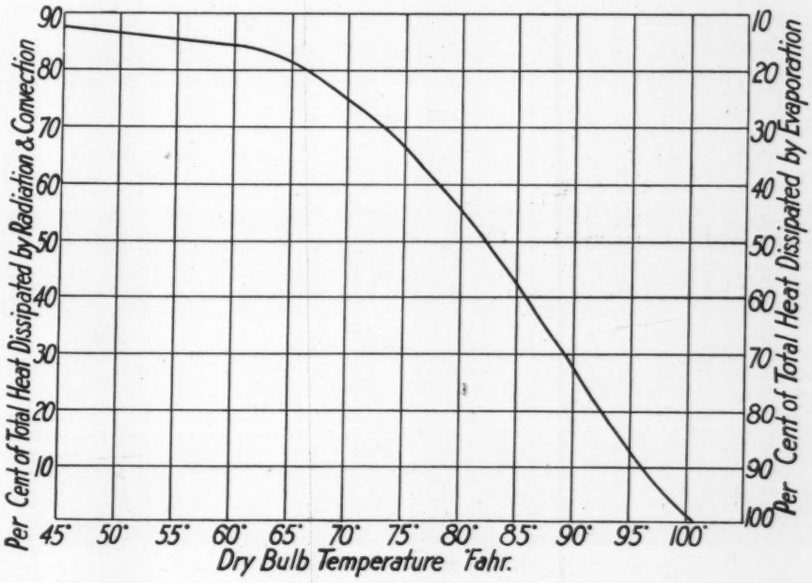


FIG. 7B. RELATIONSHIP BETWEEN SENSIBLE AND LATENT HEAT LOSSES FOR DIFFERENT DRY BULB TEMPERATURES

Economical Thickness of Heat Insulation

By R. H. Heilman

THE selection of the correct thickness of heat insulation to apply to a given case has always been of great interest to the engineer.

In spite of the fact that many articles have been written on the economical thickness of heat insulation there are still a great many engineers who do not understand the principles involved and who are unwilling to attempt to determine the correct thickness to apply to their own particular installation.

The engineer should not be criticized for the position he takes in this case, for the determination of the economic thickness of insulation for a number of different pipe sizes and operating at different pressures ordinarily requires a tremendous amount of calculation, even if one thoroughly understands the proper methods to use and is familiar with the short cuts that are possible in making the calculations.

Factors in Determining Economic Thickness

The principles involved in determining the economic size of machinery, or thickness of insulation are identical. The desired result in either case is the maximum net saving of money for any given condition.

In determining the economic thickness of heat insulation, two methods are usually used.

One method is to choose the thickness which gives the lowest annual operating expenses and the other is to choose the thickness so that the last increment of thick-

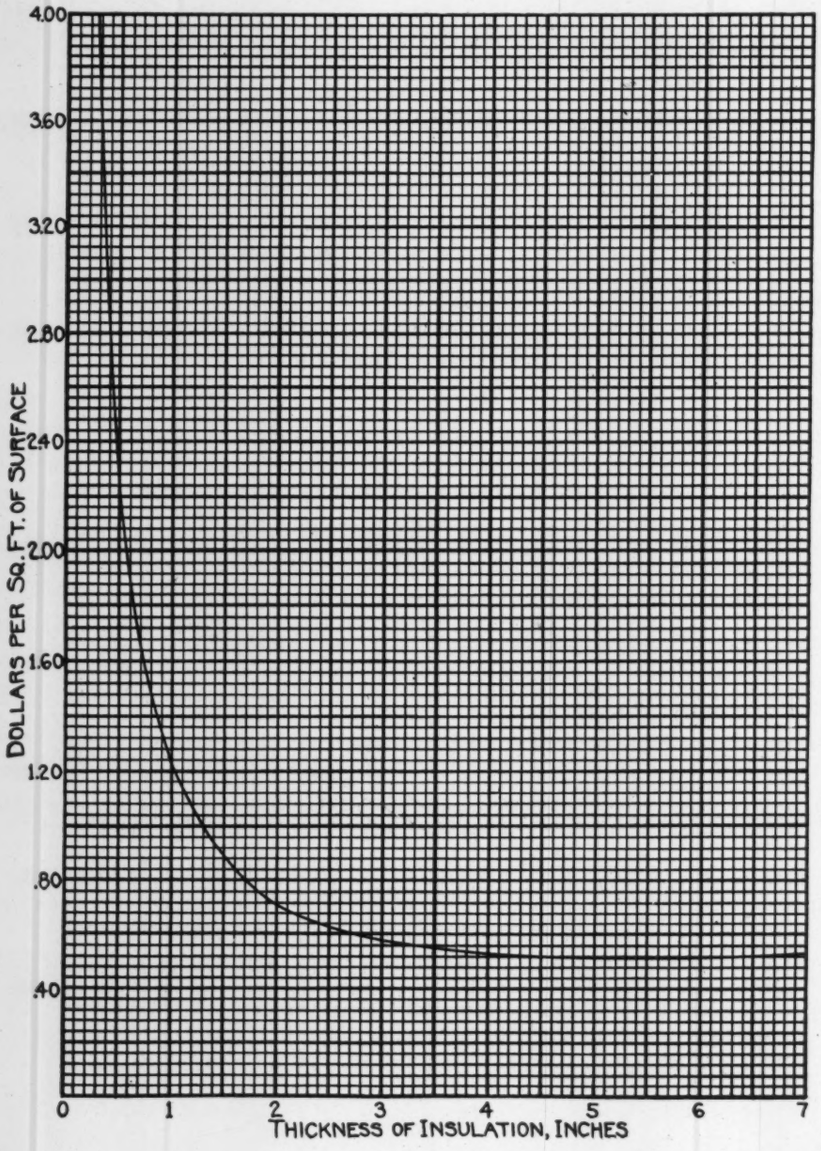


FIG. 1. TOTAL ANNUAL COSTS FOR FLAT SURFACE

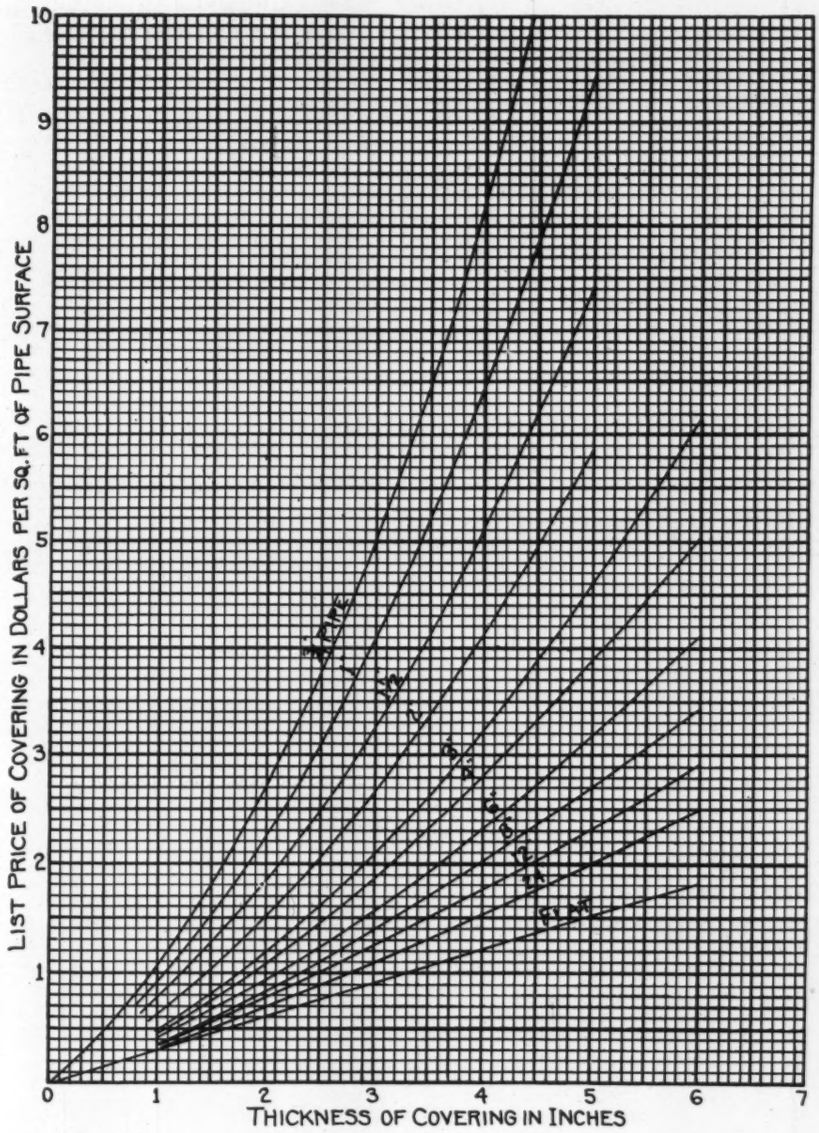


FIG. 2. LIST PRICE OF COVERINGS

ness pays a required minimum return on the additional cost. Both methods should give exactly the same results; however, the latter method, in the writer's opinion, is the better one to use, as will be explained later.

In the first case, it is necessary to know the total yearly operating expenses of the covered line. The total yearly operating expenses consist of the annual cost of the heat lost through the covered line and the annual fixed charges on the cost of the covering itself.

The following example illustrates the various steps required for the determination of the proper thickness to apply for a given case using the first method.

Example: Determine the economical thickness of covering to apply to a flat surface at a temperature of 770 deg. fahr. with air temperature of 70 deg. fahr. The covering to be used has a mean conductivity value of 0.6 B.t.u. per hr. per sq. ft. per inch per deg. fahr. temperature difference, and costs list minus 20 per cent applied. The annual fixed charges including interest, depreciation, repairs, etc., are 20 per cent. Value of heat, 40 cents per 1,000,000 B.t.u.

The heat loss through various thickness of covering at a temperature difference of 700 deg. fahr. is first determined by the method outlined by the writer in HEATING, PIPING AND AIR CONDITIONING, July, 1929. In the case of flat surfaces the equation used by the writer for the surface effect is $t_a = \frac{272.5q}{q+295}$, in which t_a = temperature difference between outer surface of insulation and surrounding air deg. fahr.; q = total B.t.u. loss per hour per square foot of outer surface.

The heat loss through a 2 in. thick covering, for example, for the assumed conditions is found to be 179.1 B.t.u. per sq. ft. per hr. The value of this heat loss per sq. ft. per year with steam at 40 cents per 1,000,000

$$\text{B.t.u.} = \frac{179.1 \times 0.40 \times 8760}{1{,}000{,}000} = \$0.61.$$

Taking the list price of heat insulation as 30 cents per board foot, the applied cost of the 2 inch thickness, at list minus 20 per cent = \$0.48 per sq. ft. The annual fixed charges were taken at 20 per cent so that the total annual fixed charges = $0.48 \times 0.20 = \$0.096$ per sq. ft.

Adding to this the annual value of heat loss of $0.61, we have 0.61 + 0.096 = $0.706 per sq. ft. per year total charges.

In order to determine the proper thickness to use, it is necessary to determine the total yearly charges for several different thicknesses so as to ascertain which thickness gives the lowest total yearly charges.

Further calculations indicate that the total yearly charges on 3-in., 4-in., 5-in., 6-in. and 7-in. thicknesses are respectively 0.576, 0.525, 0.510, 0.516 and 0.533.

It is not necessary to calculate the losses through so many thicknesses but in order to show the general trend of the curve the total yearly costs for various thicknesses from ½-in. to 7-in. inclusive have been calculated and the results shown in Fig. 1.

An inspection of this graph indicates that the lowest point on the curve or the economical thickness is approximately 5.5 inches.

In determining economic thicknesses by this method quite frequently a curve for annual operating expense is drawn and then another curve for the annual fixed charges is drawn and the values of the ordinates of the curves are added together for thicknesses varying by about 0.1 inches. The lowest value obtained corresponds to the correct thickness to use, as it will always be found that the yearly charges will increase by moving to either the right or the left of the proper thickness.

To assist any one who might wish to determine economical thickness by actual calculation, the approximate list price in dollars per sq. ft. for various thicknesses of pipe covering and for various pipe sizes is given in Fig. 2. In some cases the list price for various thicknesses of pipe covering does not follow a smooth curve, therefore, the curves shown in Fig. 2 are only approximately correct for some of the thicknesses and pipe sizes.

Economic Thickness from Return on the Last Increment of Thickness

In the second method the yearly operating expenses corresponding to the value of the heat lost through the insulation are determined as before for various thicknesses of covering. From these values the yearly saving of each additional thickness of insulation over the previous thickness is determined. For instance, the value of heat loss for the 2-inch thickness is $0.61, and for the 3-inch thickness $0.432, the saving by the additional thickness is therefore 0.610—0.432 or $0.178. The saving of a 4-inch thickness over the 3-inch thickness is $0.099. Each additional increment of thickness results in a saving over the previous one, but the amount becomes smaller as each increment of thickness is increased.

The applied cost of each additional thickness is then determined. For example, the applied cost of the 2-inch thickness is $0.48 and the 3-inch thickness is $0.72, or the additional cost of the 3-inch thickness is $0.24. The return on the additional investment of the 3-inch over the 2-inch is therefore $\frac{0.178}{0.24} \times 100 = 74.2$ per cent.

The return on the additional investment of the 4-inch thickness over the 3-inch $= \frac{0.099}{0.24} \times 100 = 41.2$ per cent.

As the increments of thickness are added the return on the additional investment becomes smaller and smaller. Since 20 per cent fixed charges have been assumed for this example, the additional increment of thickness which will show a return of 20 per cent gives the economic thickness for this particular case.

In Fig. 3 the per cent return on various increments of thickness up to 6 inches is shown.

An inspection of this graph shows that the economic thickness for this example is 5.65 inches, as any thickness above 5.65 inches will return less than the required minimum of 20 per cent on the investment.

An inspection of Figs. 1 and 3 shows the advantage, as explained, of the second method in determining accurately the economical thickness of insulation for a given case. In Fig. 3 it is not necessary to search for the low point of the curve as in Fig. 1, but it is only necessary to observe where the curve crosses the 20 per cent line. Also, in Fig. 1, it is seen that there is very little difference in the total annual operating charges for either the 4, 5 or 6-inch thickness of insulation. The actual saving of the 5-inch thickness over the 4-inch is 1.5 cents per sq. ft.; while the saving of the 5-inch over the 6-inch is 0.6 cents. Most engineers would therefore quickly say, we will choose the 4-inch thickness, since it gives practically the same economy as the 5- or 6-inch thickness.

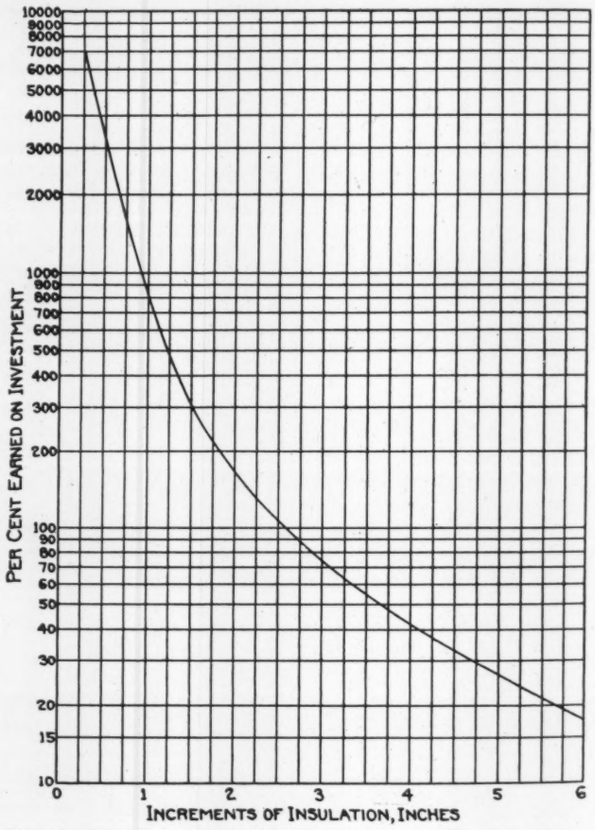


FIG. 3. PER CENT EARNINGS ON ADDITIONAL THICKNESS OF INSULATION

This view is practically always taken by engineers

when discussing papers on economical thickness of insulation which are presented before the various engineering societies.

However, the writer feels that this is not the best way to analyze the problem, but that the analysis should be made on a percentage basis as is outlined in the second method. While a saving of 1.5 cents may not seem to be of any importance, when this is distributed over a period of years and for large areas, the saving may amount to considerably more than expected. Also, the fact is often neglected that the saving of 1.5 cents results after a fixed charge of 20 per cent has been deducted.

The true value of any investment, regardless of how large or small, is best determined on a percentage basis. For example, the return on the investment of the additional thickness of 5 inches over 4 inches is 26.2 per cent. We, therefore, see that the apparent 1.5 cents saving results in a gross return of 26.2 per cent. If we allow 10 per cent for depreciation and 5 per cent for repairs and insurance, we will still have a return of 11.2 per cent on the investment. This would ordinarily be considered a good investment even if the saving was apparently only 1.5 cents per sq. ft.

In working out the two cases given above, one assumption was made that would not be used in practice. For example, the applied cost of the covering was assumed at list minus 20 per cent, regardless of the thickness of the covering. Actually the applied cost for different thicknesses is not proportional to the list price, because the labor cost of application, etc., will be nearly the same for, say, a 1-inch thickness as for a 2-inch thickness. This fact should be kept in mind, especially when the economical thickness is fairly high, as, for instance, in the examples given previously. The applied cost for each thickness should be determined as accurately as possible before making the calculations. This is necessary because the calculations must be based upon the additional applied cost of one thickness as compared with the nearest commercial thickness below it.

Chart for Determining Economic Thickness

In Fig. 4 is shown a chart which can be used in the determination of economical thickness of insulation without the use of further calculations.

An economic thickness chart practically identical to that shown in Fig. 4 was first presented by the late L. B. McMillan in a paper read before the *National District Heating Association*, 1927. In presenting Fig. 4, the writer wishes to give Mr. McMillan credit for first presenting this type of chart.

In calculating the values for his chart, Mr. McMillan eliminated the calculations for outside surface resistances by deducting 0.3 inch from the economical thickness to allow for the insulating value of the surface resistances.

However, the writer has re-calculated the entire chart, using the surface resistance formula for cylinders and flat surfaces as given in the July issue of HEATING, PIPING AND AIR CONDITIONING. A comparison of the two charts indicates that there is little difference in the results obtained from the two methods.

This chart is not extremely accurate for all conditions as it will be found that supplementary calculations will not always check exactly with the values obtained from the chart. This is due to the fact that the economical thicknesses as calculated for various conditions will not always fall on smooth curves, but that the thicknesses will sometimes vary slightly from what would be expected. However, the chart shown in Fig. 4 is sufficiently accurate for all practical purposes and its use will enable the engineer to determine quickly the economical thickness of insulation for most cases, with the exception of very high temperature conditions such as are met in the refractory field. The chart allows for practically all the variables met with in engineering practice. The variables included in the chart with a brief discussion of each one are as follows:

1. *Hours of operation per year.* Where the plant is not in operation the entire year, the actual number of hours per year that the steam is in the lines should be used.

2. *Value of heat in dollars per 1,000,000 B.t.u.* Actual value of heat at the point where the insulation is to be installed.

3. *Temperature difference.* Temperature of the metal surface over which the insulation is to be applied minus temperature of the surrounding air. For saturated steam the metal temperature will usually be practically the same as the steam temperature. For superheated steam a metal temperature about 15 deg. fahr. lower than the steam temperature is often assumed.

4. *Conductivity of insulation.* Values given in the chart are B.t.u. per square foot per degree temperature difference per 1 in. thick per hour.

It is a well known fact that the conductivity of heat insulating materials increases as the temperature is increased. Conductivity values are therefore usually plotted as a function of the mean temperature of the insulation. The engineer is usually supplied with such a chart by the manufacturer. Having this chart with the conductivities of various materials plotted as a function of mean temperature, it is rather hard for the engineer to know what conductivity value to use in the chart, as he has no means of knowing the mean temperature of the insulation. The mean temperature of the insulation is the mean of the temperatures of the inner and outer surfaces of the insulation.

The inner surface temperature can be taken the same as the metal temperature but the outer surface temperature is quite variable and depends upon the resistance of the outer surface of the insulation. This in turn depends upon the conductivity and thickness of the insulation, pipe diameter, temperature, etc.

Ordinarily the temperature drop from the outer surface of the insulation to the surrounding air due to this surface resistance may vary from, say, 10 to 100 deg.

As a rough approximation to the mean temperature, it is suggested that values between these two limits be used, depending upon the conditions. An assumption of an outer surface temperature 50 to 75 deg. above surrounding air will usually be satisfactory.

However, for some insulating materials that have very steep conductivity curves, or in which the conductivity increases very rapidly as the temperature increases, an error of 50 deg. fahr. in the mean temperature will cause an error of as much as 0.05 B.t.u. in the conductivity.

If greater accuracy than the above method is desired,

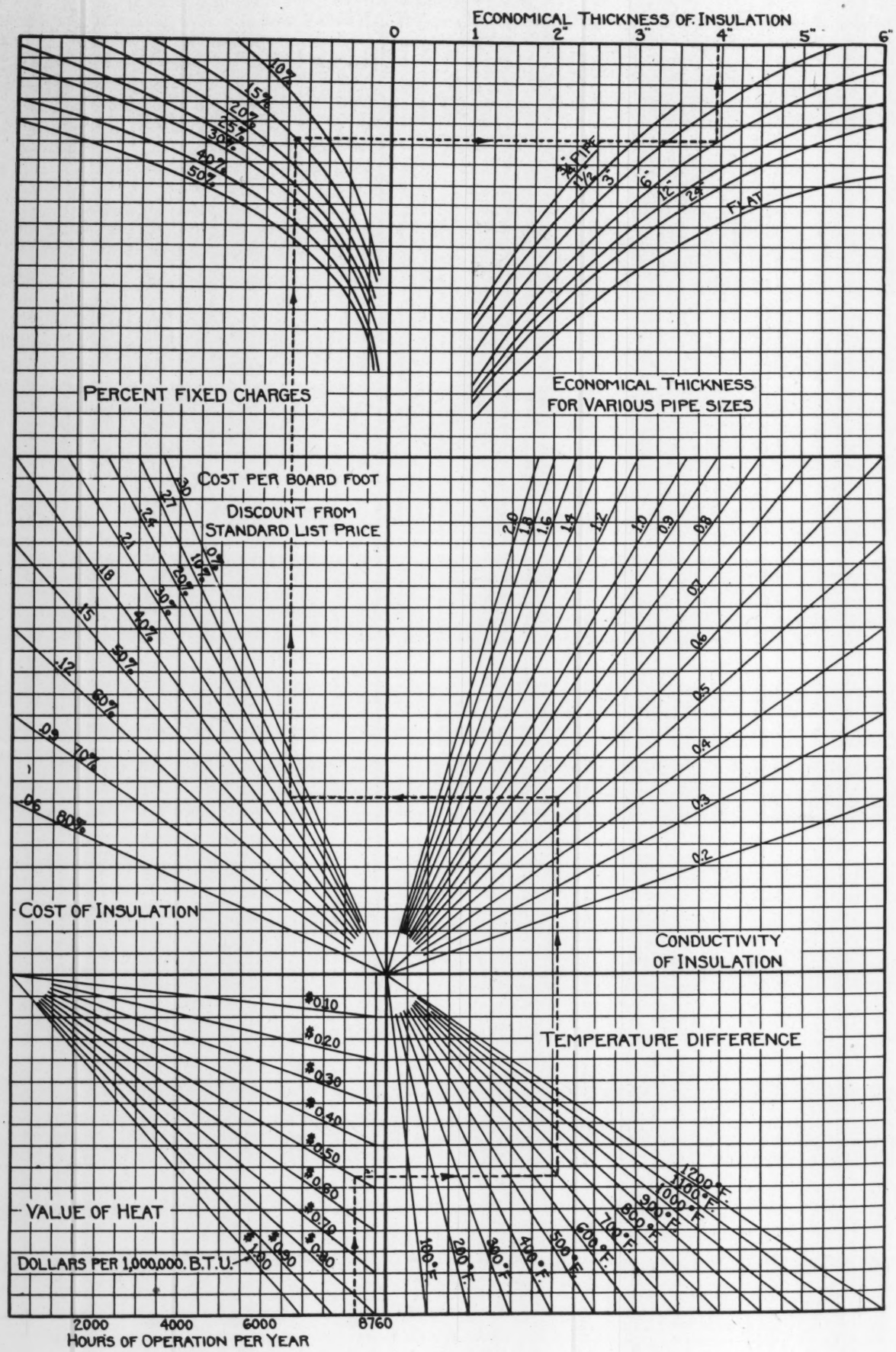


Fig. 4. Economical Thickness of Insulation

the mean temperature can be determined accurately by the following method.

This method consists in substituting in the general heat loss equation and the equation for the canvas surface loss and solving for t_d, the temperature drop from the outer surface to the surrounding air. In this case we have

$$q = \frac{k(t_1 - t_3 - t_d)}{r_2 \log_e \frac{r_2}{r_1}} = \frac{564\ t_d}{D^{0.19}(272.5 - t_d)}$$

in which

k = conductivity of insulation
t_1 = temperature of inner surface of insulation
t_3 = temperature of surrounding air
t_d = temperature difference between outer surface and surrounding air
r_1 = radius of inner surface of insulation
r_2 = radius of outer surface of insulation
D = diameter of outer surface of insulation

The above equations can be solved for t_d by transposing the equations into a quadratic equation. However, the resulting equation is very long and it is much easier to substitute first in the equations all the known values and then simplify and transpose into the quadratic.

An example will best serve to illustrate the method.

A 6-in. pipe at a temperature of 470 deg. fahr. is to be insulated. The room temperature is 70 deg. fahr. With an insulating material having a conductivity value of 0.5 at a mean temperature of approximately 300 deg. fahr., the economical thickness is found to be about 2 inches for this particular installation. Required the mean temperature of the 2 inch thick insulation.

Table 1 gives values of $r_2 \log_e \frac{r_2}{r_1}$ for various thickness of covering on various sized pipes. The value of $r_2 \log_e \frac{r_2}{r_1}$ for a 2 inch thick covering on a 6-in. pipe = 2.507.

Values of $\frac{564}{D^{0.19}}$ can be found in Table 2, page 188, July issue this journal. For the 10.6 inch outer diameter, the value is found to be 360.

We can now substitute in the equations given above, or

$$q = \frac{.5(400 - t_d)}{2.507} = \frac{360\ t_d}{272.5 - t_d}$$

Simplifying

$$0.199\ t^2_d - 494.45\ t_d + 21790 = 0$$

Solving the quadratic

$$t_d = \frac{494.45 - \sqrt{494.45^2 - 4 \times .199 \times 21790}}{2 \times .199}$$

$$= 44.8 \text{ deg. fahr.}$$

The mean temperature is $\frac{470 + 70 + 44.8}{2}$ or 292.4 deg. fahr. The conductivity value of 0.5 should then be checked with the conductivity curve at a mean temperature of 292.4 deg. fahr., and if the check is not close enough the calculations can be repeated.

5. Cost of insulation. Values are given in per cent discount from the standard list price and also per board foot. The applied cost should always be used, remembering that the applied cost for additional thickness is less than would be expected from the list prices of the thicker coverings.

6. Per cent annual fixed charges. The value to use depends upon the return required on the investment, the depreciation, etc. For power plant work a value of twelve to fifteen per cent is generally used. When the insula-

Table 1

VALUES OF $r_2 \log_e (r_2/r_1)$ FOR COVERINGS OF VARIOUS THICKNESSES ON DIFFERENT SIZED PIPES

Pipe size, in.	r_1	Thickness of Covering 1 in.		1½ in.		2 in.		2½ in.		3 in.		4 in.	
		r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$
½	0.420	1.420	1.730	1.920	2.920	2.420	4.235	2.920	5.660	3.420	7.165	4.420	10.400
¾	0.525	1.525	1.625	2.025	2.733	2.525	3.965	3.025	5.294	3.525	6.710	4.525	9.760
1	0.657	1.657	1.532	2.157	2.563	2.657	3.710	3.157	4.950	3.657	6.278	4.657	9.110
1¼	0.830	1.830	1.448	2.330	2.402	2.830	3.470	3.330	4.622	3.830	5.855	4.830	8.500
1½	0.950	1.950	1.400	2.450	2.322	2.950	3.340	3.450	4.445	3.950	5.620	4.950	8.170
2	1.187	2.187	1.335	2.687	2.194	3.187	3.145	3.687	4.177	4.187	5.262	5.187	7.658
2½	1.437	2.437	1.288	2.937	2.062	3.437	2.996	3.937	3.965	4.437	5.000	5.437	7.230
3	1.750	2.750	1.240	3.250	2.010	3.750	2.856	4.250	3.770	4.750	4.740	5.750	6.840
3½	2.000	3.000	1.216	3.500	1.959	4.000	2.770	4.500	3.647	5.000	4.580	6.000	6.590
4	2.250	3.250	1.190	3.750	1.913	4.250	2.700	4.750	3.598	5.250	4.450	6.250	6.385
4½	2.500	3.500	1.177	4.000	1.880	4.500	2.640	5.000	3.465	5.500	4.340	6.500	6.210
5	2.781	3.781	1.161	4.281	1.845	4.781	2.588	5.281	3.383	5.781	4.226	6.781	6.043
6	3.312	4.312	1.138	4.812	1.795	5.312	2.507	5.812	3.266	6.312	4.071	7.312	5.790
7	3.812	4.812	1.117	5.312	1.761	5.812	2.448	6.312	3.190	6.812	3.955	7.812	5.626
8	4.312	5.312	1.101	5.812	1.732	6.312	2.405	6.812	3.114	7.312	3.860	8.312	5.450
9	4.812	5.812	1.109	6.312	1.710	6.812	2.370	7.312	3.060	7.812	3.790	8.812	5.335
10	5.375	6.375	1.085	6.875	1.694	7.375	2.334	7.875	3.007	8.375	3.707	9.375	5.220
12	6.370	7.375	1.073	7.875	1.660	8.375	2.285	8.875	2.932	9.375	3.614	10.375	5.060
14	7.000	8.000	1.067	8.500	1.654	9.000	2.262	9.500	2.900	10.000	3.566	11.000	4.986
16	8.000	9.000	1.059	9.500	1.630	10.000	2.236	10.500	2.852	11.000	3.500	12.000	4.860
18	9.000	10.000	1.048	10.500	1.620	11.000	2.203	11.500	2.817	12.000	3.455	13.000	4.795

tion is installed for a comparatively short time, a higher rate should be used.

7. Shape of surface. Curves are given for different pipe sizes and for a flat surface. When the desired pipe size is not shown in the chart, the economical thickness can be obtained by interpolation.

The dotted line on the chart illustrates its use in solving a typical example.

Use for Underground Construction

This chart has been calculated for above-ground conditions; however, it can be used for underground construction where the pipes are installed in tunnels, provided the air temperature of the tunnel is known. In this case, the chart is used the same as for above ground work, the temperature difference to use in the chart is the difference between the pipe temperature and the tunnel temperature. Where the underground insulation is placed in conduits or where it is not surrounded by air, it is difficult to determine accurately the economical thickness to use, due to the uncertainty of the insulating value of the ground. In these cases, a thickness of one-half inch less than for the corresponding above-ground conditions is often used.

Capitalized Value of Ten B.t.u. per Hour

In selecting heat insulation for a particular installation, the engineer is naturally interested in obtaining the maximum insulating value per dollar spent. The manufacturer usually supplies tables showing heat losses through various thicknesses of insulation on various pipe sizes.

However, since the values are usually given in B.t.u. per sq. ft. per hr., a saving of ten or fifteen B.t.u. is quite often considered of little importance, as the engineer cannot readily visualize this saving in dollars and cents.

To enable one to transpose B.t.u.'s to dollars and cents and to enable the engineer to determine readily how much more he can justly pay for one thickness over another, or for one type over another in order to save, say, ten B.t.u., Table 2 has been prepared.

Table 2 shows the capitalized value of ten B.t.u. per hr. Since the monetary value of ten B.t.u. depends upon the value of the heat saved, the table is based upon steam costs of from $0.15 to $0.60 per 1,000,000 B.t.u. The per 1,000,000 B.t.u. costs of from $0.15 to $0.60 have been transposed to costs of various fuels as follows: Cost of steam per 1,000 lb. with heat content of 1,200 B.t.u. per lb., cost of coal per ton of 13,000 B.t.u. per lb., cost of gas per 1,000 cu. ft. of 1,000 B.t.u. per cu. ft., cost of gas per 1,000 cu. ft. of 600 B.t.u. per cu. ft., and cost of fuel oil per gallon of 152,000 B.t.u. per gallon. The costs are for the actual value of the heat at the point of application of the insulation. The known cost of fuel fired must be divided by the efficiency of the equipment.

The table shows the permissible investment to obtain gross returns of from 6 to 100 per cent for a saving of ten B.t.u. per hr. for the various values of fuel. To obtain the permissible investment for any other B.t.u. saving, it is only necessary to multiply the values given in Table 2 by the actual B.t.u. saving and divide the result by 10.

The values given in the table are for gross returns and do not include depreciation on the insulation. However, to take care of this item, it is necessary to add the depreciation to the per cent return desired on the capital investment and use this value to determine the permissible investment.

Examples

The value of this table is best illustrated by examples. If we take for instance the example in the former part of the paper in which the economical thickness of insulation was determined for a flat surface operating at a temperature difference of 700 deg. fahr. with a steam cost of $0.40 per 1,000,000 B.t.u., we find that the heat loss through the 5-in. thickness is 77.3 B.t.u. per sq. ft. per hr., while the loss through the 6-in. thickness is

Table 2

CAPITALIZED VALUE OF 10 B. T. U. PER HOUR

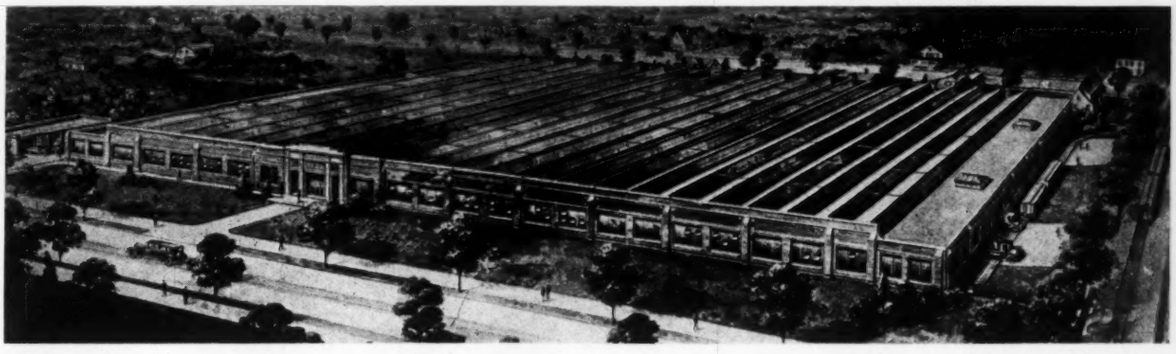
Cost of fuel per 1,000,000 B. T. U.	0.15	0.20	0.25	0.30	0.40	0.60
Value per year	0.0131	0.0175	0.0219	0.0263	0.0351	0.0536
Cost of steam per 1,000 lbs. 1200 B. T. U. per lb	0.18	0.24	0.30	0.36	0.48	0.72
Cost of coal per ton, 13,000 B. T. U. per lb	3.90	5.20	6.50	7.80	10.40	15.60
Cost of gas per 1,000 cu. ft. 1,000 B.T.U. per cu. ft	0.15	0.20	0.25	0.30	0.40	0.60
Cost of gas per 1,000 cu. ft. 600 B. T. U. per cu. ft	0.25	0.33	0.42	0.50	0.67	1.00
Cost of oil per gallon 152,000 B. T. U. per gal	0.023	0.030	0.038	0.046	0.061	0.091
Per cent gross return	Permissible Investment, Dollars					
6	0.218	0.292	0.365	0.438	0.585	0.877
8	0.164	0.219	0.274	0.329	0.439	0.658
10	0.131	0.175	0.219	0.263	0.351	0.526
12	0.109	0.146	0.182	0.219	0.292	0.438
14	0.094	0.125	0.156	0.188	0.251	0.376
16	0.082	0.109	0.137	0.164	0.219	0.329
18	0.073	0.097	0.122	0.146	0.195	0.292
20	0.066	0.083	0.110	0.131	0.175	0.263
25	0.052	0.070	0.088	0.105	0.141	0.210
30	0.044	0.058	0.073	0.088	0.117	0.175
35	0.037	0.050	0.063	0.075	0.100	0.150
40	0.033	0.044	0.055	0.066	0.088	0.131
50	0.026	0.035	0.044	0.053	0.070	0.105
75	0.017	0.023	0.029	0.035	0.047	0.070
100	0.013	0.018	0.022	0.026	0.035	0.053

65.1 B.t.u. per sq. ft. per hr., which gives a saving of 12.2 B.t.u. per sq. ft. per hr. by using the 6-in. thickness over the 5-in. Referring to Table II, we see that to save ten B.t.u. with a gross return of 20 per cent, we are justified in spending $0.175. However, with an applied cost of list minus 20 per cent, the additional cost is $0.24 per sq. ft. We are therefore not justified in selecting the 6-in. thickness. The economical thickness is therefore between 5 and 6 inches.

The table is especially useful in determining the value of one insulation over another when the heat loss through a given thickness of the various insulations is known. For instance, if *A* shows a saving of fifteen B.t.u. over *B* with a fuel cost of $0.40 per 1,000,000 B.t.u. and a desired return of 15 per cent on the investment with 10 per cent depreciation on the insulation, we find from the table that we can spend $.141 $\times \frac{15}{10}$ or $0.21 per sq. ft. more for *A* than *B* and realize 15 per cent on the additional investment.

It should also be remembered that the economic thickness of the covering having the highest insulating value is less than the covering having a lower insulating value even if the two coverings sold for the same price. With a lower price on the covering having the lower insulating value, the economic thickness of the cheaper covering is still greater than it would be if the two coverings were sold at the same price.

A future article will contain a chart for obtaining heat losses through various thicknesses of covering from the conductivity curve of the insulation.



THE LELAND ELECTRIC COMPANY PLANT AT DAYTON, OHIO, IS HEATED BY FIFTY-THREE UNITS VENTING DIRECTLY THROUGH THE SAW-TOOTH ROOF. THIS ELIMINATES THE SMOKESTACK

Direct Fired Units Heat Large Factory

By Everett S. Buck

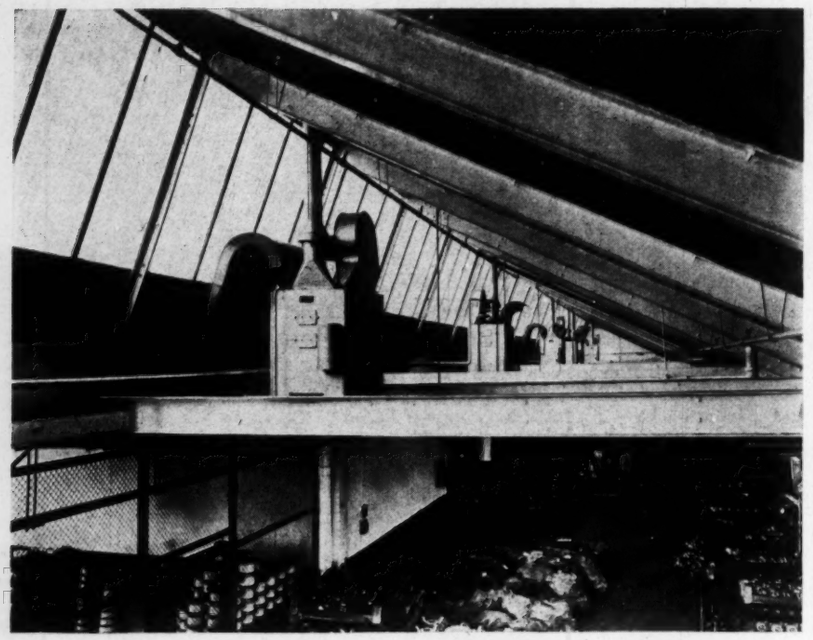
SIMPLY described, the heating system recently installed in the large plant of the Leland Electric Company consists of 53 direct gas fired, electric fan driven, and controlled unit heaters, located at strategic points in the roof trusses throughout the plant. Absolutely no floor space is sacrificed; each area of forty by forty feet has individual temperature control.

All burned gases are vented directly through the sawtooth roof. Venting the burned gases gives us our first loss of heat, so let us see what this amounts to. A good rule-of-thumb factor is that one cubic foot of air is required for every 100 B.t.u.'s of gas burned Suppose we require 25 per cent of excess air to burn one cubic foot of gas having a value of 1,000 B.t.u.'s, and we raise this air temperature to a point 100 deg. fahr. higher than when it entered for combustion.

A rough computation of the sensible heat loss per 1,000 B.t.u.'s will then be $12.5 \times 100 \times .0182 = 22.7$ B.t.u., or roughly 2¼ per cent of our total heat. The actual performance of these units in the Leland plant is better than that computed above, and if we were to assume that with complete combustion and a loss in sensible heat of about 2¼ per cent that all of the rest of the heat would be available, we should have nearly 98 per cent efficiency.

This is not the case, however. Almost every fuel contains hydrogen. If this hydrogen is burned completely, it associates during combustion with the oxygen in the air to form H_2O, and this is included with the flue gases in the simple form of water vapor. The surprising thing about this water vapor is that its heat content is, on the average, more than 10 per cent of the total value of the gases, and this heat is recoverable only if the vapor is condensed as steam is condensed in a radiator. This water vapor is not condensable as is ordinary steam at 212 deg. fahr., but, following Dalton's law of the partial pressure of gases, is condensable with theoretically perfect combustion at about 135 deg. fahr. for the average gas.

Due to the construction and provision for complete counter flow of air and gases, the units in the Leland plant condense a major portion of this water vapor which exists in the burned gases and recovers its



THE UNITS ARE MOUNTED ON THE ROOF TRUSSES. EACH FORTY BY FORTY FOOT AREA IN THE LARGE PLANT HAS INDIVIDUAL TEMPERATURE CONTROL

latent heat, although the gases are being vented at a temperature considerably above the actual dew point.

The units handle the condensate that results in the following manner: Once set, production of water vapor per unit will be constant, since it is a function of combustion, dependent on the temperature of the air recirculated through the unit. The condensate recovered by the unit is driven through a limestone filter to neutralize the acid and is then passed through a water seal to be evaporated by the air which is being swiftly forced through the unit, resulting in an increase in humidity.

One temperature is maintained during the day and another during the night, this operation being controlled by clock-operated electric switches.

Apparently, the only regular attention that need be given is to oil the motors once a year, light the pilots in the fall and turn them off in the spring, and wind the clocks weekly.

No Control of Humidity

There is no attempt at room control of humidity which necessarily varies with the amount of infiltration, but there is a constant addition of moisture, and a close relation between need for moisture and its production. If the system is designed for zero, during 30 deg. weather it will run approximately two-fifths of the time and add moisture when it is running. During zero weather when the need for moisture is greater it will run from four-fifths to nearly 100 per cent of the time and add moisture for humidity in proportional rate. Under normal conditions the result will be a slight deficiency in moisture when below-freezing temperatures prevail outdoors, and the resulting indoor relative humidity is usually just below the point where condensate on single sash begins to become a nuisance.

For industries requiring more humidity the deficiency can be made up by an automatically controlled humidifying system.

Terminal Tower Building, Cleveland

IN THE TERMINAL TOWER BUILDING STEAM IS SUPPLIED BY THE POWER PLANT OF THE ELECTRIC ILLUMINATING COMPANY. WINDOW VENTILATION IS USED THROUGHOUT THE OFFICE SUITES, BUT THE KITCHENS AND DINING ROOMS ARE VENTILATED MECHANICALLY. THERE ARE 20 FAN UNITS OF VARIOUS SIZES IN THE BUILDING

Heating and Ventilating Cleveland Schools

By Arthur E. Hutchinson*

HEATING and ventilating are major requirements in the operation of public schools. Conditions over an extended area may be so similar that studies in one school may be interesting or useful to all other schools in that area. For instance there is not much difference in the requirements in the cities of Omaha, Denver, Chicago, Detroit, Buffalo, Pittsburgh and Cleveland. Their temperature ranges are very similar. This is particularly true in states that have uniform ventilation laws and which have definite heating or ventilating requirements for schools.

There is probably no building that presents so many problems in heating and ventilating as are found in schools, for in the school building we have as many ideas volunteered regarding what is thought to be the proper condition as there are principals and teachers. When to their ideas are added those of the supervisors and others who visit the building occasionally, as well as the parents of many of the pupils, the problems gather weight. Open windows, open doors, unit heaters, central fans, direct radiation, indirect radiation, conditioned air, unconditioned air,—these are a few of the simpler suggestions offered as solutions of the heating and ventilating question. It sound simple, but it isn't.

For this reason it is believed that the procedure and investigations conducted in heating and ventilating as well as the experiences of the bureau of operation of the Cleveland board of education may be interesting to similar bureaus in other cities and to engineers in general.

Design of the school heating and ventilating plant is of great importance and only with suitable design and intelligent operation can satisfactory performance be had. Lacking proper design the operator will have difficulty in achieving economical operation. The design should be simple, yet have all the equipment necessary for good heating and ventilating practice.

Much improvement has been made in the heating and ventilating plants for city schools. Sometimes there has been a period of retrogression but the trend over a period of years has been upward. One may see evidence of this progress by inspecting the heating and ventilating equipment of city schools, and it probably would be rather astonishing to many to observe the advancement that has been made.

There is, however, a great variety of equipment used. In the 150 schools of Cleveland for instance, one may find practically every kind of heating system. There is the old drum furnace, the blast system heated by furnaces, the blast system heated by steam coils, the split system heated by combination furnaces and steam boilers, the split system heated by steam only, direct radiation—single pipe, direct radiation—two pipe, the vacuum system, gravity system, etc., all of which average a fair degree of satisfaction.

A Standard Heating Plant Needed

It appeared, however, that there should be a standard heating plant for moderate-sized schools and in 1924 a committee was organized for the purpose of standardizing on such a plant. This committee was headed by George M. Hopkinson, now architect of the Cleveland board of education who at that time was superintendent of buildings. Other members of the committee were representatives of the architect's office and representatives of the operating bureau, the former in charge of the design of school buildings and the latter responsible for the proper operation of school equipment.

It was decided to send out questionnaires, one on the subject of boilers and one covering heating plants, including the fans, unit heaters, etc. These questionnaires were sent to twenty-three cities ranging in size from the largest to the smaller ones.

In these questionnaires much consideration was given to the selection of boiler plant equipment. The committee at length decided upon the following standardized plant: two horizontal return tubular boilers, of 150 h.p. each, set in battery, not less than six ft. from floor to the bottom of the shell, designed for 100 lb. with a working pressure of 50 lb. per sq. in. with reduction to 2 lb. in the house lines and a chimney of sufficient capacity to carry both boilers.

Up to this time the Ohio laws limited the pressure to 35 lb. within the four walls of a school. However, owing to the use of steam-driven stokers and other auxiliaries, 50 lb. working pressure was considered more suitable and consequently all schools that were being built in Cleveland for several years prior to 1924 were designed with a detached or semi-detached boiler room.

In the winter of 1924-25, the legislature revised

* Asst. to chief of the bureau of operation.

the building code and a new requirement was that no high-pressure boiler would be permitted to operate within the four walls of a school building. The old construction of the phrase "low pressure," meant 15 lb. The question then arose, what is a low pressure boiler plant? The law being retroactive could put out of commission the majority of the heating plants in Cleveland, as well as those in other cities which were already operating at 35 lb. as well as those which wished to operate with the higher pressure, unless in a detached boiler room. A committee from Cleveland appeared before the board of building standards for an interpretation of the revised code and it was agreed that for the purposes of the code any heating boiler carrying under 51 lb. was a low pressure plant.

THE OLD—

Stoker Equipment

At the time, practically all the boiler plants being built were equipped with either side or front overfeed stokers. Probably every city has its own method of operation for its school buildings, but owing to the fact that Cleveland, in nearly all cases, lets its fires go out overnight instead of banking them, it was decided that interlocking, shaking and dumping flat grates would be installed with a furnace of such design as to be as nearly smoke-eliminating as possible. It was felt that a plant of this kind would be simple, easy of maintenance and low as to first cost.

Smoke Elimination

This furnace has not been changed in its fundamentals. We added to it, however, an inverted arch back of bridge wall and have installed steam jets of a patented type in both side walls and just above the grate. These furnaces, with the added arch, have proved to be very efficient in smoke elimination and have helped firing conditions. Contrary to the general belief that jets draw heavily on steam, in the course of a test made on one of our boiler plants it was found that the total consumption average was only 27 lb. per hr.

In connection with this type of furnace, smoke observers are being installed giving the operator very definite knowledge of his fire conditions and assisting very materially in the elimination of smoke.

Much difference of opinion exists here in relation to the merits of metering devices in heating plants. We installed a simple indicating meter of the mercury column type which has given good satisfaction. Meters of this type are open to the disadvantage of not leaving a record and are subject to the personal equation in reading them, but they give us a very definite slant on what the boilers are doing.

Suction soot blowers are used and are giving excellent satisfaction. Some hesitancy was at first expressed relative to them as it was felt that with only 50 lb. working pressure the cleaning area would be limited, although satisfactory operation was guaranteed at 40 lb. A very close watch was kept of the inside of the tubes at the combustion chamber in order to note whether the soot was being thoroughly removed and in no instance did we find evidence of soot deposit remaining after a thorough blowing. Boiler front doors, breeching connections, etc., should be tight, otherwise there will be an exceedingly disagreeable leakage of soot into the boiler room.

Feed Water Heating

In two plants we inject live steam into a receiver between the vacuum pump and the boiler feed pump. In both instances there have been positive savings in fuel. In one plant where four boilers were in use ordinarily, the operators were able to get along with only three last year. In nine buildings where we have open feed water heaters, we show less fuel cost than in similar buildings where we are operating without them and operating conditions are better.

Three Buildings Alike Used for Making Comparative Tests

In view of the fact that the three buildings to be built were exactly alike, it was agreed that this would provide us with what would be comparable to a laboratory in which to do our own experimenting and develop for ourselves information for future guidance.

In one of the buildings, there are being installed two underfeed stokers of the unit type with the conventional side dump grates, small self-contained blast fans, etc., while in the other two are being installed flat shaking and dumping grates with steam jets for smoke elimination.

In the 31 room standard buildings the direct and indirect radiation are each fed by separate low-pressure mains, each controlled from the high-pressure main by a shut-off valve, pressure-reducing valve and three-valve bypass so that either system can be used independently of the other if desired or in case of emergency.

A coal vault of approximately 400 tons capacity is built directly in front of the boilers with the top at ground level and a drop side charging coal truck is part of the standard equipment. No coal or ash conveyor is installed in these buildings. A flue gas pyrometer and a two-tube draft gage for uptake and over fire draft are part of the standard equipment.

Choose Central Fan System

After taking various points into consideration and after consulting the returned questionnaires, the committee decided upon a properly designed central fan system. There were many differences of opinion, but eventually the committee decided upon a split system using sufficient direct radiation to take care of wall losses and infiltration, the indirect radiation to take care of air changes to comply with the code.

The 31 room standard buildings are "T" shape and exactly symmetrical about a center line through the boiler and fan rooms, with an auditorium on the first floor and a play room or small gymnasium on the second floor of the stem of the "T." There are three supply fans in the basement, two for class rooms and one for auditorium and play room. This arrangement lends itself perfectly to either day or night activity, as it is only necessary to operate the fan serving that part of the building being used. As the greater part of the night activities in a building are either in the play room or auditorium it can be heated with a minimum of expense.

—And the New

Each of the supply fans in the basement runs in series with an exhaust fan in the attic, and while the Ohio law is to the effect that six changes of air must be admitted to the class room, instead of the thirty cubic feet per minute per pupil, as originally set up by the code, and recirculation is not allowed during school hours, all operators will acknowledge that for the purpose of heating up in the morning, recirculating is almost a necessity in large buildings.

The intake of the supply fan is on the roof, insuring a much cleaner and fresher supply of air than when taken at ground level as is the case of nearly all of the older types of buildings. The discharge of the exhaust fan is also at the roof. The two sets of fans are cross-connected and operated by air control from the basement fan room, so that the operator can, at will during the heating up period, open and close the bypass dampers, thereby recirculating one hundred per cent if so desired.

Recirculation as a Fuel Saver

Many will undoubtedly question the advisability of this arrangement, feeling that the temptation to use the recirculating switch at other times than when heating up, in order to save labor and fuel, is too great. However, out of fairness to our operators, we have never found one recirculating during school hours and certainly it is a fuel saver for the early morning period when it is necessary to heat up the school buildings.

All of the class room fans are opposite hand and are located in the center of the basement with the auditorium and play room fan between them. This makes an ideal arrangement from the standpoint of load distribution. The fans are all motor-driven.

The Sanitary Duct

Other cities, notably Chicago, were using the "sanitary duct," running it longitudinally with the building underneath the first floor corridor, and the committee adopted this type of main duct. These ducts are lined with glazed tile, have vacuum cleaning outlets in them, and are equipped with floor drains, allowing them to be washed out as often as desired. Thus they are kept clean and free from dust.

The risers lead directly from the duct up through the corridor wall to the rooms. There are two supply inlets and two exhaust outlets to each room, both on the corridor side of the room, the supply inlets being 8 ft. above the floor and the exhaust at floor level. There are no turns in the risers from the main duct to the elbow as it enters the room and, as the volume dampers are free opening, the risers may be brushed out or vacuum cleaned at any time, insuring complete cleanliness.

As the indirect system is for ventilation only the control is simpler than if it were used for heating as well. There is only one set of coils, part of which are controlled by a thermostat in the fresh air intake and the other by a thermostat in the duct after the air leaves the fan. This temperature is normally set at about 75 deg. fahr. and with approximately 325 ft. velocity at the grilles, we are experiencing no difficulty from drafts. The exhaust fan exhausts from two-thirds to three-quarters of the air supplied, creating plenum condition in the room at all times.

Building Is Zoned

A system of day and night control is used, the building being divided into zones, making it possible to lower the temperature in unoccupied rooms in order to conserve fuel. It is still questionable to some, however, whether in a building of this size the extra expense of this type of control can be justified, particularly in view of the fact that in most elementary buildings the fires are allowed to go out or are banked at night.

All temperature control dampers are operated from a central control board, the indicating unit giving the operator accurate knowledge at all times of his damper position. The air control compressors in duplicate are also located in the fan room where they are always under observation. Too many times small machines of this type are located in out-of-the-way places. The idea of "out of sight, out of mind" may be all right for some things, but it is poor policy for heating plant machinery.

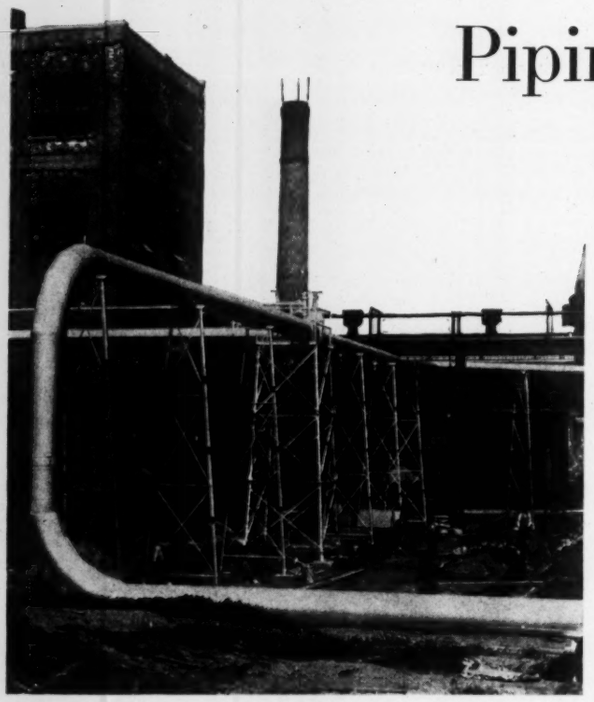
Humidity Required

Air washers were given special consideration. For several years prior to 1924 they had been installed in all buildings built, but always in the minds of many interested was the thought that humidification, particularly in the Cleveland climate, is a refinement not justified in practice if one considers the length of time a pupil is in the school room, as compared with the length of time spent outside, especially for elementary pupils.

It was agreed that for buildings in the less thickly populated districts and in the cleaner sections, air washers would not be used. However, space was left and provision made so that they can be installed.

Piping Acetylene Gas

By B. M. Conaty



Twenty-four Inch Acetylene Gas Line Rising to Steel Tower Across Railroad Tracks and Entering Plant. Valves Are on the First Trestle

AMONG the varied operations of the Shawinigan Chemicals Company is a process which requires large quantities of acetylene gas. The acid plant in which the gas is consumed is located about 3,500 ft. away from the point of generation of the gas. Floating on the line between these points is a 50,000 cubic feet gasometer, made necessary by the fact that uniform consumption of the gas is impossible. Originally the acetylene was transmitted through two lines 8 in. and 10 in. in diameter.

A large addition to the acid plant increased the gas requirements. At the same time it was noted that the losses in the 8 in. and 10 in. lines approximated 25 per cent. It is probable that these losses were almost entirely due to leaks in the pipe, the leaks in turn being traceable to the installation underground, above the frost line, with insufficient provision for expansion and contraction.

These conditions resulted in the construction of a 24 in. line with a maximum capacity of 120,000 cu. ft. per hour. Several features make this line a most unusual one.

The pressures encountered are extremely low, amounting to 10 in. of water at the generator and 5 in. at the gasometer approximately 1,000 ft. from the point of production. At the acid plant, 3,500 ft. from the generator, the minimum allowable pressure is 4 in., so that the pressure drop in the 2,500 ft. of line is limited to 1 in. of water.

Pipe Made from Plates in the Field

Under these pressure conditions standard pipe is not required and the duty on fabricated pipe is considerably greater than on steel plate. A decided economy was effected by the purchase of ¼ in. steel plate 78 in. wide and 10 ft. long. This plate was rolled into cylindrical sections in the field and welded longitudinally to produce 10 ft. sections, which in turn were welded into the line.

While there is no heat in the line, seasonal changes produce a temperature difference of approximately 150 deg. fahr., which necessitates the use of six externally-guided expansion joints. In order to use welded construction wherever possible these joints were furnished with cast steel bodies so that they could be welded in place.

The line is supported on concrete, steel or wood structures, as the conditions required, spaced on approximately 24 ft. centers. At each support a roller is installed to facilitate easy movement of the line. Fabricated steel alignment guides are used to pro-

vide the necessary rigidity and anchors are located approximately 500 ft. apart. Much of the steel work supporting the line was built on the job, using material reclaimed from previous alteration of the plant.

All joints in the line are welded, using the oxy-acetylene process, except where branch line gate valves are installed and where welding was impracticable. The light pipe wall did not permit homogeneous bends; these were welded, using a sufficient number of sections to provide long radius turns. Much of the steel support work is of welded construction.

Unique Valve Use

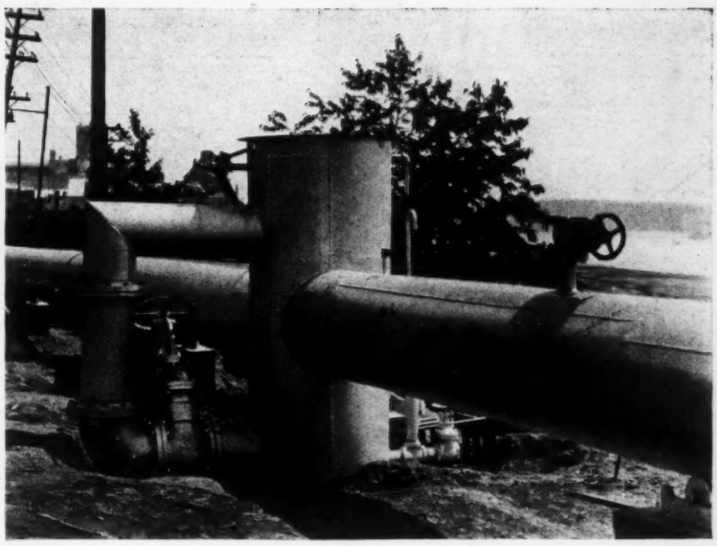
The explosive nature of acetylene gas requires extreme caution in its handling and resulted in the development and construction on the work of a unique emergency shut-off valve. This valve consists of a solenoid-operated water seal device, which can be operated from any one of several push buttons located at strategic points in the plant and along the line. In addition to the remote control feature in this valve a contact breaker is provided which closes the valve instantly when the pressure on the line falls to zero, due to explosive rupture or other causes. A vacuum breaker is also incorporated in the valve.

The nature of the service is such that no covering is required on the line. However, it is weatherproofed with one coat of aluminum primer and three coats of aluminum paint applied by spraying.

The Line Showing an Expansion Joint Housing

It is estimated that a saving of more than 20 per cent of the total cost was obtained by fabrication in the field of everything but the expansion joints, roller supports and gate valves. This, of course, is due largely to the difference in duty on steel plate and on shop fabricated or manufactured materials. In this country no such results could be obtained, for it is probable that hammer-weld pipe of the proper wall thickness or other standard material could be purchased as economically, at least, as it could be fabricated in the field.

The new line is proving a profitable investment.



Automatic Cut-Off Valve

Fidelity Trust Co. Building, Philadelphia

THE office building of the Fidelity Trust Company in Philadelphia is thirty stories high with three basements, and contains 618,000 sq. ft. of net usable area. The volume of this structure is 11,500,000 cu. ft.

Five water tube boilers supply a total of 1,609 horsepower; the working pressure is 160 lb. Anthracite rice is the fuel used. It is interesting to note that three 6,000 gal. fuel oil tanks have been installed in case future conditions make this type of fuel advisable for use in this building. A thirty hp. fan takes air from the top of the boiler and pump rooms and delivers it through the underside of the grates.

Air Conditioning an Existing Building

By Fred B. Orr

THIS installation was made in two adjoining buildings, in which there is a large area of floor space with comparatively low ceiling height. An unusually large light load emanating heat from a lamp section, and a boiler room directly below the floor in another section, caused an average heat rise of approximately 15 deg. fahr. with normal occupancy of these spaces. During the noon and evening rush hours, as the store patronage increased, there was an even greater heat rise. This additional heat from the increased number of people in the sales space was concentrated to a great extent in the main cross aisles leading from street to street. The air conditioning equipment had to be made sufficiently flexible to compensate for these uneven heat loads in the shop. The difficulty of diffusing the air properly in an installation of this type will readily be appreciated by those familiar with air conditioning problems.

The installation of air conditioning systems in existing buildings involves problems and difficulties of construction quite different from problems surrounding new buildings. At the present time almost half of the air conditioning installations are made in existing buildings. Obviously, no consideration was given in the original designs of such buildings for the future installations of so-called "cooling systems." The equipment and the distribution systems to be installed in these structures have distinct limitations both in style and in size. As an added limitation in some buildings the same general architectural treatments must be retained.

The successful solution of these problems is admirably exemplified in the recently completed air conditioning system in the retail sales and display rooms of the Commonwealth Edison Company in Chicago.

In order to absorb the heat gain in the conditioned areas, the diffusion temperature (the temperature of the entering air below the room temperature) had to be fixed at a point between 10 and 18 deg. The quantity of air to produce an equivalent cooling effect varies inversely with the diffusion temperature. After careful experiment and checking, it was found that any diffusion temperature within the possible range might cause objectionable drafts. Consequently, extreme care was taken in the design and the location of outlets. The location of grilles was governed somewhat by existing building conditions; but in general the system was designed for one row of grilles supplying a distance of not more than thirty feet in the direction of air delivery, and adjacent grilles in the row not more than fifteen feet apart. The outlets were installed with separate volume and direction adjustments, so that minor variations in volume and direction could be made in the final balance of the system. Thus a screen of air across the ceiling is obtained which produces an even and comfortable air condition in the sales spaces.

The available space in the basement provided unusually cramped quarters for the installation of cooling coils, compressor, air washer, intake, and proper sized distributing ducts. The basement of this office building is crowded with four 1,000 h.p. boilers and the incidental steam mains running through the basement that supply heat to many other loop office buildings; two 2,000 kw. rotary converters with adjacent busbars; an emergency storage battery room; and all the usual power equipment of a modern office building. Such were the extraordinary obstacles encountered in obtaining straight runs of air ducts with the proper cross-sectional area. These runs had to be constructed to connect the air conditioning equipment with the risers to the first floor, which supply the grilles in the ceiling beams.

Old Ducts Used

Air which has served its purpose in the sales space is returned to the basement equipment through outlets that originally formed a part of the old ventilating system of the Edison and Marquette buildings. These outlets and the duct system to which they are connected supplied and exhausted air from the electric shop prior to the installation of the new air conditioning equipment. They were completely altered by engineers to serve as returns for the conditioned air which has been introduced into the sales spaces.

A centrifugal supply blower with a capacity of 56,000 cu. ft. of air per minute and operating against 2 in. water gage pressure was installed. The high static pressure was necessary in order to overcome restrictions encountered in running the supply ducts through the building.

The dehumidifier washes, cools and dehumidifies the mixture of outdoor and recirculated air. The heat exchange involved in this process takes place in two stages. In the first stage the air comes into contact with atomized water in the spray chamber for a period of eight-tenths of a second. In the second stage, the air is passed for one-half of a second around direct expansion coils sprayed with water.

The pump in the Edison installation has a capacity

FIG. 1—A VIEW OF THE ELECTRIC SHOP IN THE COMMONWEALTH EDISON BUILDING, CHICAGO, SHOWING THE GRILLES IN THE PLASTER BEAMS

of 650 gallons per minute. The suction line to the pump connects with the pan under the dehumidifier. The discharge line from the pump supplies water to the two stands of mist nozzles in the spray chamber, and to the flooding nozzles over the bank of direct expansion coils.

Counter-current Principle Used

Air is cooled in the dehumidifier on a counter-current principle. The water which is discharged through the spray chamber nozzles is maintained at a temperature of 42 deg. fahr. After the warm air is treated in the spray chamber, it passes on to the direct expansion coils, where the final cooling and drying of the air is accomplished. The same 42 deg. water is also sprayed over bunker coils in the direct expansion chamber but in this stage the water and coils are at a considerably lower temperature because of the cooling effect produced by a gas maintained at 25 degrees inside the coils. The water sprayed over the direct expansion coils is cooled at a temperature considerably below the 42 deg. entering temperature, in spite of the fact that it has absorbed considerable heat from the air passing through the chamber. As a result, when the final mixture of water from the spray chamber at a temperature above 42 deg. is mixed with the water from the direct expansion chamber, there is a resultant temperature of 42 deg. fahr. at the pump suction. The process is continuous and balanced by an initial adjustment of valves controlling the volume of water to each stage, and by setting refrigerant expansion valves.

This method of heat exchange has been found by engineers to be very efficient. Losses incident to radiation through long lines of cold piping and to a separate water cooling unit outside the dehumidifier are eliminated.

Labor Organization

The organization of the erection and the direction of the erecting crew is of utmost importance in an installation of this type, where the space to be conditioned is constantly used for business purposes during the day. All work in the sales space had to be done at night; and all debris, scaffolding and materials had to be removed before the opening of the shop in the morning. At first thought, one would think that the cost of the installation would be materially increased because of the overtime work. Only one-fourth of the total labor, however, was in the sales space; and the rest represented the fabrication of ducts and erection of equipment. It was found that the mechanics tend to organize more readily than on a "straight day" job; and thus a good part of the additional cost is absorbed. In the final analysis, the additional cost was found to be less than a 5 per cent increase over "straight day" costs.

Air Cooling and Dehumidifying Equipment

The air cooling and dehumidifying equipment in the sales spaces of the Commonwealth Edison Company is of the carbonic anhydride safety type, which operates through direct expansion coils placed di-

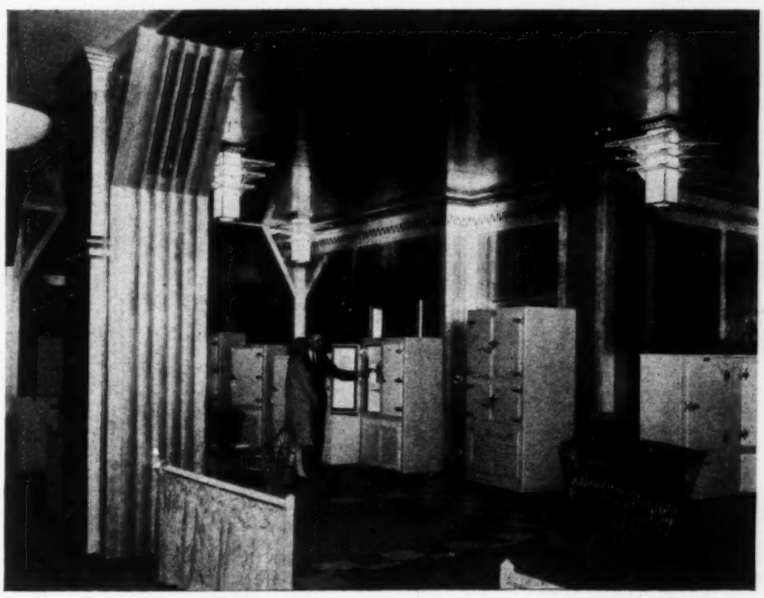


FIG. 2—THE REFRIGERATION DEPARTMENT OF THE ELECTRIC SHOP REQUIRED DECORATIVE TREATMENT FOR THE AIR SUPPLY GRILLES. THE COLUMN SHOWN IS FURRED OUT TO CONCEAL THE SUPPLY DUCT. THE OUTLET APPEARS AT THE CAP OF THE COLUMN

FIG. 3—IN INSTALLING THE AIR CONDITIONING SYSTEM, THERE WAS THE NECESSITY OF PRESERVING THE ARCHITECTURAL TREATMENT OF THE DISPLAY ROOMS. HOW WELL THIS WAS ACCOMPLISHED IS INDICATED IN THE VIEWS HERE. BELOW IS A SECTION OF THE DISPLAY ROOM BEFORE THE INSTALLATION, AND AT THE TOP IS THE SAME VIEW AFTERWARD. THE GRILLES MAY BE SEEN IN THE BEAMS

rectly in the dehumidifier, as described above. In order to fit the limited space conditions an air cooling duplex compressor unit consisting of two horizontal, single cylinder, double acting compressors with a common shaft was installed. The compressors were designed for direct connection with an engine type direct current motor. They are of heavy duty type with all moving parts completely enclosed and automatically lubricated. Each of the compressor cylinders has a stuffing box on the piston rod to confine all gas pressures in the cylinders. The stuffing boxes are equipped with a pressure oil feed system for sealing the boxes and lubricating the piston.

For driving the duplex compressor unit, a 175 h.p., 230 volt direct current compound wound motor of engine type is direct connected to the common shaft. The motor is designed to operate at a speed of 164 r.p.m. It is equipped with a push button starter complete with all necessary protection, that simplifies operation required to start and stop the large compressor unit.

Carbonic anhydride condensers used in connection with this cooling and dehumidifying plant are of the double pipe, welded type. Water flows through the inner pipe, and the carbonic anhydride gas is discharged into the annular space between the inner and outer pipes. All gas joints are electrically welded to make them gas tight. The inner pipes are connected with bolted return bends, which may readily be removed for cleaning the water tubes.

Cooling Coils Welded

The direct expansion cooling coils comprise a total of approximately 6,000 lineal feet of 1¼-in. extra heavy galvanized pipe installed directly in the dehumidifier. The cooling coils are all electrically welded into continuous lengths, so that there are no flanged joints inside the dehumidifier. The expansion and stop valves required for controlling these cooling coils are installed on the outside of the dehumidifier casing. All of the pipe stubs passing through the dehumidifier casing are equipped with water tight stuffing boxes. The entire set of cooling coils are arranged so that they are flooded with water, giving a Baudelot effect.

All of the carbonic anhydride connections between the duplex compressor unit, the condensers and the cooling coils are made with extra heavy pipe with long sweeping bends having all joints welded wherever possible. The headers used at the condenser and at the cooling coils are also of welded construction to make all pipe joints and headers gas tight.

Compactness and Silence

The entire layout of the cooling and dehumidifying equipment was carefully designed for compactness to fit the limited space, and for silent operation. Since

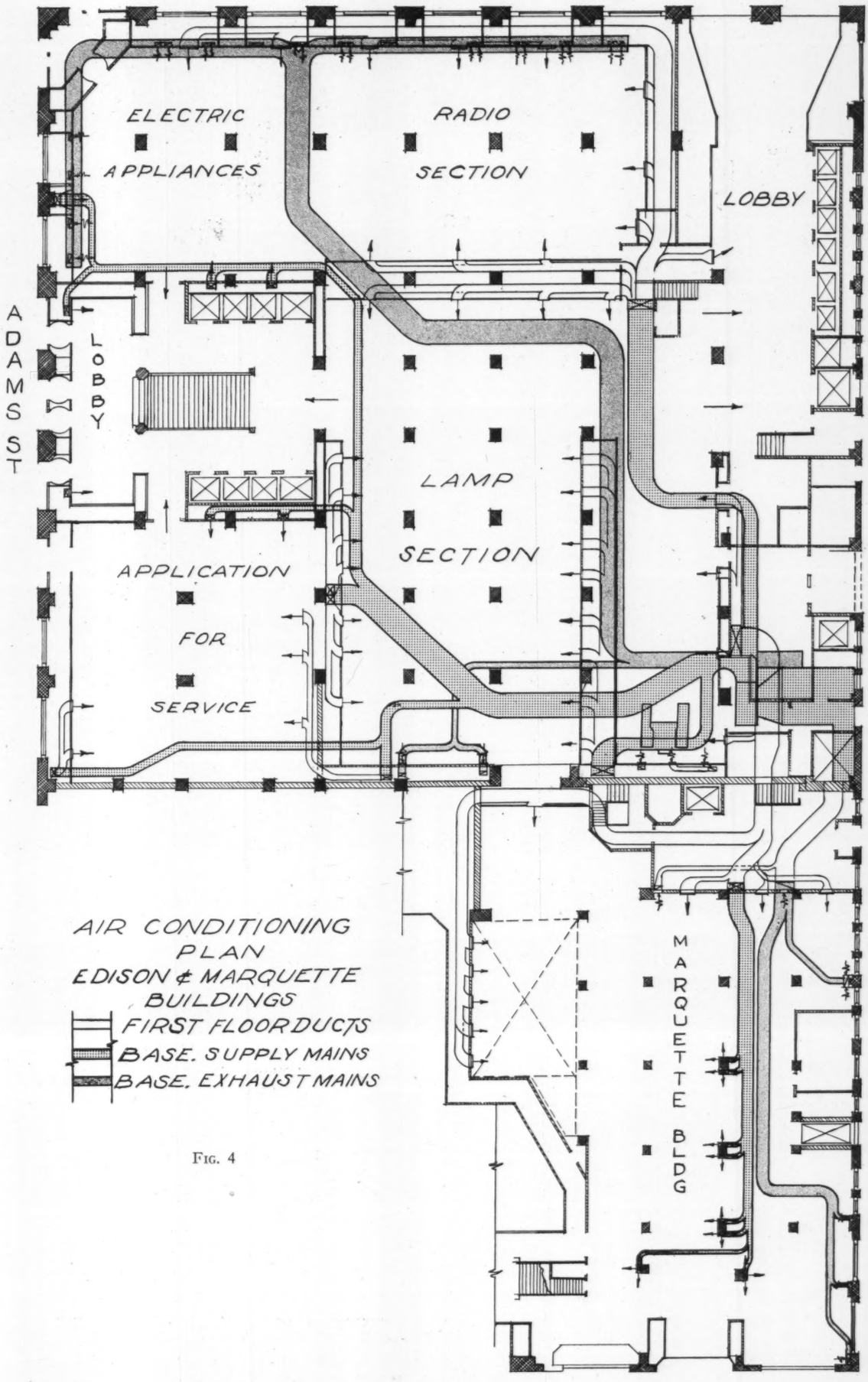

ELECTRIC
APPLIANCES
RADIO
SECTION
LOBBY
ADAMS ST
LOBBY
LAMP
SECTION
APPLICATION
FOR
SERVICE
MARQUETTE BLDG
AIR CONDITIONING
PLAN
EDISON & MARQUETTE
BUILDINGS
FIRST FLOOR DUCTS
BASE. SUPPLY MAINS
BASE. EXHAUST MAINS

FIG. 4

the equipment is located directly under the electric shop sales room, great care was taken to prevent all transmission of noise and vibration which would be objectionable in the sales room directly above. The duplex compressor unit and motor are set on a concrete foundation which is thoroughly isolated with machinery cork from the structure of the building. All pipe hangers and supports are similarly isolated.

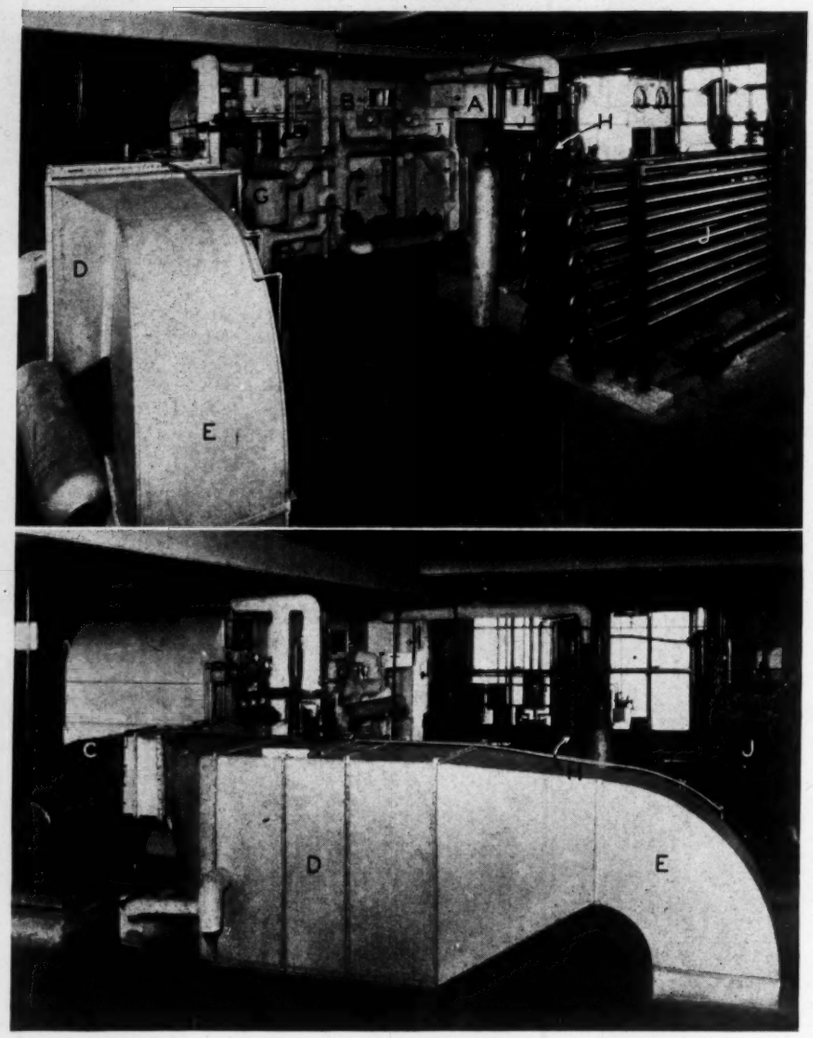


FIG. 5—EQUIPMENT FOR CONDITIONING THE AIR IN THE PRIVATE OFFICE IS IN THE ELEVATOR PENT HOUSE ON THE TWENTIETH FLOOR. EVERYTHING IN THE AIR CONDITIONING SYSTEM IS SHOWN HERE. AIR ENTERS THROUGH THE TOP HALF OF A WINDOW AND IS FILTERED BY A FIBER MAT FILTER ENCLOSED IN THE HOUSING *A*. IT THEN PASSES TO THE DEHUMIDIFIER SHOWN AT *B* OR THROUGH A BY-PASS IMMEDIATELY BACK OF THE DEHUMIDIFIER TO THE SUPPLY BLOWER *C*. THE BLOWER DISCHARGES THE AIR THROUGH HEATING COILS AND SOUND ABSORBING CHAMBER *D* INTO THE AIR DUCT *E* WHICH LEADS TO THE OFFICE. THE TANK *F* CONTAINS SUBMERGED DIRECT EXPANSION COILS TO COOL THE DEHUMIDIFIER SPRAY WATER FOR SUMMER OPERATION. THE INDIRECT HEATER *G* HEATS THE SPRAY WATER FOR WINTER OPERATION. THE ICE MACHINE INSTALLATION, INCLUDING THE COMPRESSOR *H* AND CONDENSER *J*, IS ENTIRELY AUTOMATIC AS IS THE REST OF THE EQUIPMENT

Conditioning an Office

The conditioning of air in a private office on the seventeenth floor of the Edison building presented slightly different problems. In this office no perceptible movement of air was allowed at the "breathing line" although the usual air circulation was required. The air outlets in the office had to be concealed so that the design and the decoration of the room would in no way be changed. On casual inspection no visible means for incoming and outgoing air can be found. Closer examination, however, reveals that the air is introduced and exhausted through an ornamental cove close to the ceiling. Except for the thermostat, there is absolutely no change in the appearance of the room.

The system for this office is automatically controlled to produce a predetermined room temperature and humidity. A switch in the adjoining office starts the process of air conditioning. The operation of the equipment once that it has been started is entirely automatic.

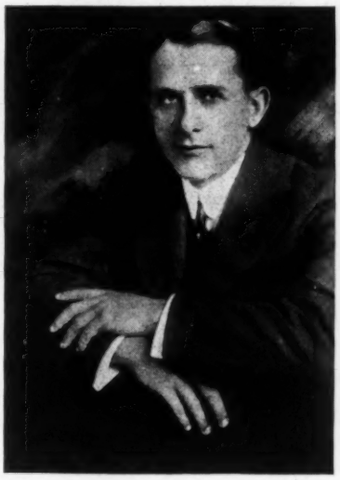
It is now generally accepted that humidity has as much, if not more, effect on human comfort than has temperature. This installation offers impressive proofs. As originally designed, the capacity in the volume of air supplied and exhausted, the heating effect, and the cooling and dehumidification effect were considerably oversized. Any reasonable condition could be produced within the room by adjustment in the operation of the equipment. In some air conditioning installations, the desired result is a noticeable contrast between outside and inside conditions, for the obvious psychological effect upon people. In this instance the desired condition was one which would provide for the continued comfort of an individual occupying the private office although capacity is provided for thirty people when present at meetings. At first a noticeable difference between outdoor and indoor temperature was maintained but there were some complaints. Finally, a much more satisfactory temperature and humidity was obtained in the room by adjusting the equipment to produce a comparatively low humidity with not much depression in the dry bulb temperature. Such was the case, for example, on a day when the outside temperature was 85 deg. with a normal wet bulb temperature, when a 79 deg. dry bulb and 38 to 40 per cent relative humidity was maintained.

Many expressions of satisfaction have been made on the condition produced as a result of these installations. The trend for many other developments of science providing for added human comfort has been from a luxury to a necessity. Air conditioning is following that trend.

Welding Practice and Standards

By John H. Zink

THE correct application of the welded joint to our piping installations, and particularly low pressure heating installations, is definitely understood by the majority of our contractors and is appreciated as an important construction method because of its economy, strength and efficiency. From an engineering standpoint the properly welded pipe joint meets every requirement as to delivery and performance. The development and production of welding fittings by our material manufacturers have been prompt and satisfactory. Thus, these three branches of our industry have been equally alert and progressive in establishing the welded pipe joint as a method of installation.



Pipe welding, however, did not reach this high position in our industry without experiencing instances of indiscriminate use and the creation of some unsound individualistic practices. Our engineering authorities, quick to understand the resulting injury, have unified their efforts and through the welding committee of the Heating and Piping Contractors National Association will define and publish in manual form our code of standards, a manual which, in addition, will be a complete treatise on the art of welding steel and wrought iron pipe.

The fact that this code has not as yet been published is not an excuse for improper pipe welding. There have always been available to us the engineering knowledge and constructive assistance necessary to permit our pipe welding installations to be made with efficiency and proper control. Those contractors who follow these accepted standards and have the vision to see the welding torch as a tool of this trade, deplore its abuse in that it tends to establish unnecessary resistance on the part of our designing engineers as to its requirement in their piping specifications. Our standards are supported by experience, research investigation and exacting tests.

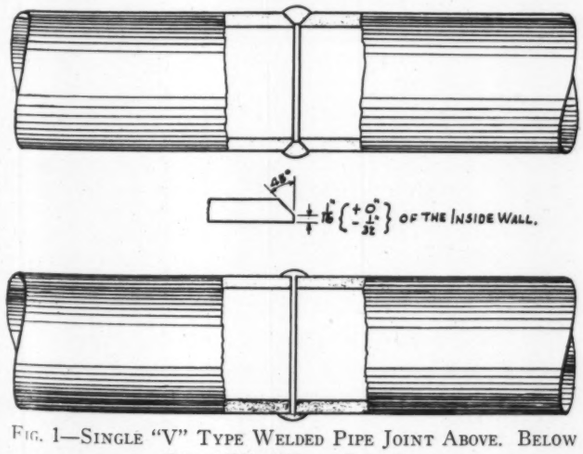


FIG. 1—SINGLE "V" TYPE WELDED PIPE JOINT ABOVE. BELOW —BUTT WELD TYPE PIPE JOINT

Publication of individualistic practices and welding methods in other countries which are contrary to those accepted by the engineering authorities of our industry only aggravate and make difficult our work of standardization. Even though the purpose of such articles is strictly educational, there is the possibility of their being misunderstood. An example is the article in the July issue of "Heating, Piping and Air Conditioning" on "Welding as Applied to Heating Installations in Central Europe" by Mr. Konrad Meier, eminent European authority.

My discussion, therefore, while primarily intended to sustain confidence in our present accepted welding standards for our installation requirements, makes direct reference to Mr. Meier's article for comparative purposes, to point out why we differ and to show wherein I cannot concur in some of the conclusions of the European authorities mentioned.

An illustration in Mr. Meier's article, shows pipe ends of joints that were welded with indifferent or poor workmanship. I assume they have been made with the butt joint method as later on in Mr. Meier's article it is stated that the "V" type welded joint is no longer considered necessary for reliable work and the butt joint is preferred and is satisfactory.* It is regrettable that Mr. Meier did not support this conclusion with the engineering data that prompted its adoption, as our experience has been that a satisfactory pipe weld with the butt joint method is an exception and not the rule.

Fig. 1 will illustrate to the reader the two types of welded joints in question.

The single "V" type weld will be insisted upon as our standard and represents no additional cost in the preparation and delivery of the pipe. Beveling the short length of pipe in the field by hand is admitted to be somewhat expensive but the development of the combination cutting and beveling machine adequately answers this question.

The only other factor that needs consideration is the difference in wall thickness of European pipe as against American pipe. While the European pipe has less wall thickness, this difference as I understand it is nominal, and if the thinner wall were used on our pipe, we would still insist upon the single "V" type welded joint. It is known that some

* The paragraph in Mr. Meier's article to which this refers is as follows:—"In Central Europe, as a rule, the butt joint is preferred and satisfactory, if sufficient space is left between contact surfaces for the material to enter. The "V" joint requires more preparation and is no longer considered necessary for reliable work."

of our pipe manufacturers at the present time have under consideration the reduction in wall thickness of their pipe and this change is directly made possible by pipe welding.

It is recognized that there are two factors of expansion to be considered as a result of applying the intense heat of the welding torch to the pipe ends; namely, longitudinally, which tends to close the spacing, and circumferentially, which tends to increase the diameter of the pipe. It is also recognized that contraction in cooling is severe and the confined internal heat must be equalized and distributed to the pipe wall in order to prevent misalignment. These factors are maximum with the butt joint and consequently spacing, even if supported with "tacks," places an unreasonable responsibility on the operator for proper penetration and fusion. Inability to determine or obtain correct welding temperatures, loss of flux control and proper consideration of the effect of the welding heat to the pipe metal are the principal contributing factors to an unsatisfactory weld.

Our experience has shown that the single "V" type welded joint meets all of these conditions of pipe welding more satisfactorily than does the butt joint in that it enables the operator to conveniently control every part of the welding operation including the elimination of "icicles" on the inner pipe wall. It enables us to weld all sizes of pipe with no special technique required for the smaller sizes.

Consideration must also be given to the quality welding rod used inasmuch as a specification welding rod is just as essential as the technique employed. If welds which can withstand any stress other than a deliberately destructive one are to be obtained, a good quality welding rod intelligently applied produces welds that are ductile and fibrous with resulting maximum strength.

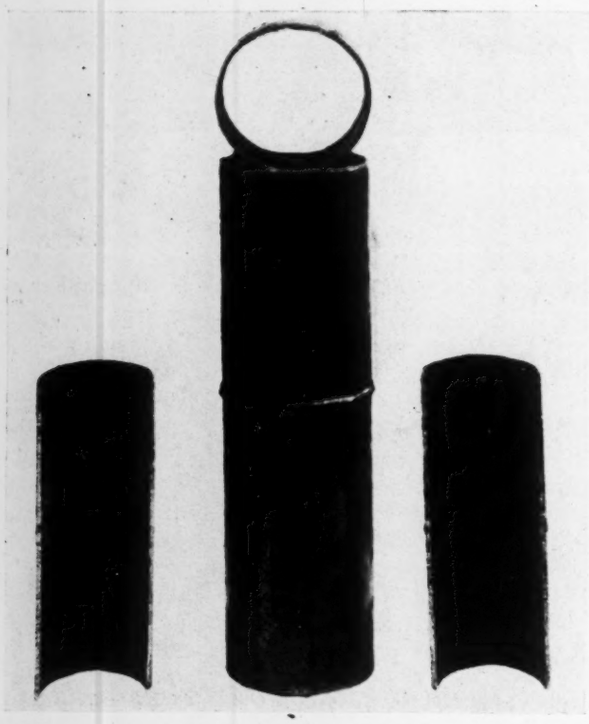


FIG. 2—TYPICAL EXAMPLES OF THE SINGLE "V" TYPE WELDED JOINT APPLIED TO PIPE WITH SECTIONS CUT AWAY

It is customary for us to guarantee the single "V" type weld to be equal to the pipe strength whereas the butt weld pipe joint gives a strength average below fifty per cent of that of the pipe. I have many coupon tests to support this conclusion, and to avoid submitting a mass of scientific data, reference is here made to the published and comprehensive report of the sub-committee on pipe joints of the American Gas Association, illustrating and describing tests of various types of welded joints. The results obtained are conclusive justification for our adoption of the single "V" type welded joint as our pipe standard.

Fig. 2 is the average result of the use of the single "V" type welded joint applied to our piping. It will be noted that maximum strength is obtained by proper fusion and that it is unnecessary for penetration to extend beyond the inner pipe wall.

Also in Mr. Meier's article there is an illustration of welded reducers made by recessing the smaller pipe end into the larger pipe end, with end views showing obstructions. Under no circumstances would our recognized practice in this country permit this fitting to be made in the described manner. For the readers' information I am showing in Fig. 3 two designs of this fitting, one being made by swaging

FIG. 3—ONE—REDUCER MADE BY SWAGING IN THE FIELD. TWO—REDUCER AS A MANUFACTURED FITTING

the larger size pipe to the diameter of the smaller in the shop or field, and the second a reducing fitting offered at reasonable prices by material manufacturers.

Either of these fittings satisfactorily meets our requirements as to cost and quality. It will be of interest to know that within reasonable time the manufactured reducing fitting will be available in types eccentric as well as concentric to meet every requirement of flow. It should be appreciated that the intended design of all of our welding fittings will be to enable us to place them in position by means of a pipe weld. This is both desirable and economical.

Mr. Meier also shows in his article an illustration of ready-made steel pipe bends welded on. I assume the

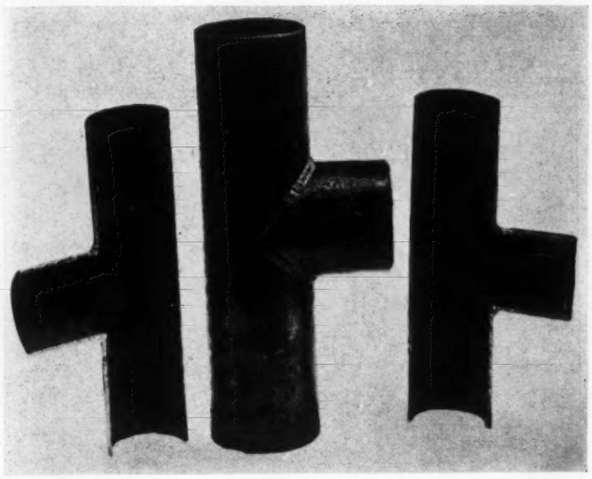


FIG. 4—WELDED TEE WITH SECTIONS CUT AWAY TO SHOW DESIGN AND WORKMANSHIP

notation on the caption of this illustration "opportunities for clogging" refers to the resulting weld rather than the design of the fitting. I concur in Mr. Meier's conclusion that the ready-made pipe bend offers more resistance to flow due to the roughing of the inner surface through the process of bending, but the illustrated fittings are similar to the ones we are using and they are not ready-made as I interpret the expression. They are scientifically manufactured from seamless tubing, have a smooth and uniform wall thickness, and the shorter radius enables us to meet practically every installation problem in the field. Their use eliminates many fittings and pipe bends. The strength of this fitting is equal to the pipe, its satisfactory use is widespread, and, in my opinion, will be encouraged.

Another of Mr. Meier's illustrations shows a branch tee or inner section weld, in which the interior view illustrates the result of poor workmanship. Fig. 4 illustrates one of our branch tees made in the field with section view showing the resulting conditions on the inside of the pipe wall. It speaks for itself and is average.

I recognize the necessity for correct design in any attempt to make this fitting and wherever possible it should be designed with the assistance of available templates. The intersection weld demands the use of the single "V" type joint if results are to be obtained similar to those shown in Fig. 4. A fitting thus made is perfectly satisfactory as to resistance to flow and strength.

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

"Outdoor Dust," by H. C. Murphy. Page 212, issue of July, 1929

YOUR July 1929 number contains an article on "Outdoor Dust" by Mr. H. C. Murphy, vice president of the Reed Air Filter Co., Louisville, Ky. This article has two tables, the first of which shows the average decrease in sunshine for various cities. In this table Mr. Murphy states that the decrease for Chicago was 12 per cent.

Mr. Murphy states in his article that he obtains his figures on decrease in per cent of possible sunshine by comparing the sunshine reports of the U. S. Weather Bureau for the three winter months (December, January and February) of 1926 with the corresponding months in 1916. His figures do not seem to be correct for Chicago as shown by the following tables:

	Per Cent		*Per Cent*
January, 1916	41	January, 1926	48
February, 1916	52	February, 1926	33
December, 1916	55	December, 1926	34
Average	49.33	Average	38.33

Decrease 11

$$\frac{11}{49.33} = 22.3\%$$

If we compare the figures for the winters of 1915-16 and 1925-26 we obtain the following results:

	Per Cent		*Per Cent*
December, 1915	34	December, 1925	44
January, 1916	41	January, 1926	48
February, 1916	52	February, 1926	33
Average	42.33	Average	41.67

Decrease 0.66

$$\frac{0.66}{42.33} = 1.57\%$$

Average for 1916..62 per cent — Average for 1926..50 per cent

Decrease 12

$$\frac{12}{62} = 18.1\%$$

It will be seen from the tables that the sunshine for the year 1926 established an all-time low figure for Chicago, while that for 1916 is one per cent within the highest figures for the last 15 years. If the writer had compared 1915 and 1928 the results would have shown an increase of about six per cent for the year and about 18 per cent for the three winter months.

I am enclosing two sheets, which are a copy of the official figures of the U. S. Weather Bureau station at Chicago. The only calculations which are mine are those for the twenty-year average from 1894-1913 inclusive, and the fifteen-year average from 1914-1928 inclusive.

Arnold H. Kegel,
Commissioner of Health, City of Chicago.

A Letter from Mr. Murphy

My article was prepared last year and in arriving at my figures I took the latest sunshine data which was then available on the 24 cities covered by my dust sur-

U. S. Department of Agriculture, Weather Bureau.
Per Cent of Possible Sunshine.

YEAR	January	February	March	April	May	June	July	August	September	October	November	December	Average
1894	57	57	52	64	67	71	85	73	70	43	37	45	60
5	46	67	68	68	73	83	82	79	86	77	41	28	66
6	35	55	65	63	78	84	66	82	56	63	36	28	59
7	38	31	50	49	62	52	59	54	71	72	32	30	52
8	48	44	57	54	57	50	76	63	58	34	49	51	55
9	49	42	32	65	61	71	66	75	54	51	38	37	53
1900	37	47	41	61	56	63	60	60	57	66	40	38	52
1	40	67	36	67	50	69	77	56	67	69	52	40	58
2	61	58	56	67	65	58	59	67	48	67	41	33	57
3	45	55	52	60	68	75	79	68	68	70	56	47	62
4	43	47	44	61	65	73	69	74	56	59	62	33	57
5	38	53	58	66	67	76	69	66	63	62	57	53	61
6	35	58	41	70	74	72	75	65	68	45	39	30	56
7	32	58	53	43	55	61	62	67	53	59	61	29	53
8	56	36	48	61	50	79	77	82	85	70	59	43	62
9	33	30	40	46	68	55	68	72	68	60	47	46	53
10	34	52	84	56	55	80	78	68	60	71	45	42	60
11	28	44	63	56	79	79	75	76	57	47	41	42	57
12	42	46	55	65	69	80	65	57	67	71	63	51	61
13	42	66	60	69	64	82	75	67	65	54	43	41	61
Average not including: 1911, 12, 13	43	50	52	60	63	69	71	69	68	64	49	41	58
Highest Mean	61	67	84	70	79	84	85	82	86	77	62	53	66
Lowest Mean	28	30	32	43	50	50	59	54	48	34	32	28	52
Range	33	37	52	27	29	34	26	28	38	43	30	25	14
1914	25	59	55	55	74	77	80	73	78	60	67	42	62
15	40	40	56	74	45	61	50	51	55	65	52	34	52
16	41	52	49	50	58	67	95	86	66	66	53	55	62
17	64	58	55	53	60	60	71	73	66	33	49	44	58
18	58	60	77	53	68	73	74	65	55	50	52	30	60
19	60	47	60	48	66	71	86	81	67	51	50	36	60
1920	48	44	63	50	75	75	78	72	71	69	45	44	63
21	57	38	49	62	70	69	88	72	70	62	35	33	59
22	73	56	49	59	65	82	75	75	70	71	30	44	62
23	44	47	55	61	69	73	75	64	60	61	48	47	59
24	59	41	31	57	62	68	76	62	63	84	58	49	59
25	48	51	71	64	70	73	68	75	56	31	54	44	59
26	48	33	45	59	72	75	70	60	37	42	29	34	50
27	35	54	46	53	45	70	72	66	60	62	28	46	53
28	49	48	58	57	72	49	66	69	73	51	34	39	55
29	46	52	51	51	56	67	68	.....	.....	.....	.....	.....	.....

1894-1913 incl. Twenty year annual average = 57.75.
1914-1928 Fifteen year annual average = 58.20 Increase 0.8%.

vey. This consisted of the bound volumes of the United States Weather Bureau containing the Monthly Meteorological Summaries, Form 1030, which gives the "Average Percentage of Sunshine" for each month for the 24 cities in question. For the three months under consideration in 1926 this gives 37 per cent as the "Average Percentage of Sunshine" for Chicago. The same source gives the average for these months ten years earlier, in 1916, as 49 per cent. Subtracting the "Average Percentage of Sunshine" in 1926 from the 1916 average gives the figure used in my table. The same method was used in arriving at the sunshine decrease for each of the other cities.

While the percentage of sunshine in Chicago in 1926 was perhaps lower than the average, this was not the case in a number of the other cities—for instance, New York, Philadelphia, Baltimore, etc. Also I believe it will be found that the year 1916 did not have the highest average percentage of sunshine. I find, for instance, in 1922 the average for Chicago for the three winter months was 58 as compared with 49 in 1916.

An attempt was made in the article to drive home the economical loss due to smoke in our city life but as the survey covered the whole country and has to do with an exceedingly variable subject, the possibilities of controversy and contention are many.

I appreciate Dr. Kegel's comments and the spirit of his letter. I have been intensely interested for a number of years in the movement for smoke elimination and I am in hearty accord with the present campaign to "clean up the air" in Chicago.

I am further convinced that when the general public awakens to the staggering economical cost of smoke in men and money, they will take whatever steps are necessary to eliminate it.

H. C. Murphy.

Heating·Piping
and Air Conditioning

1895

American Society of Heating and Ventilating Engineers

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NOVEMBER 1929

Material in this Section prepared by Publication Committee: A. C. Willard, *Chairman*; N. W. Downes, and W. A. Rowe. : A. V. Hutchinson, Manager of Publications

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Power from Process and Space Heating Steam

By L. A. Harding[1], Buffalo, N. Y.
MEMBER

THE necessity for the utilization of recoverable wastes and what are termed by-products in practically every industry becomes increasingly apparent each year and is primarily due to competition and plant location.

The utilization of a particular recoverable waste in one plant may not, owing to the relatively advantageous location of the plant, be either practical or profitable, whereas, a competitor, on the other hand, finds that it is not only profitable but an economic necessity. The mere fact that a waste is recoverable is no criterion that its recovery is either desirable or a profitable venture. A new process is rarely, if ever, so perfect that it is not susceptible to economic improvement by recovering some form of waste whether it be in the form of a material product or a reduction in the fuel or electric power bill. It so happens in one industry, at least, that the price of the principal product is largely determined by the market value of the by-product from the process.

A given amount of money spent for improvement of the process, the utilization of more up-to-date machinery, etc., may show a better return on the investment than the same amount spent on the equipment required for a waste recovery. It is not generally a simple matter to estimate with any great degree of accuracy the equivalent money value from the recovery of a by-product, the amount and value of the product saved or recovered being subject for various reasons to considerable fluctuation.

Electric power rates in various parts of the country show a great variation. Low fuel cost and a high power rate are naturally a combination that would appear to be the most desirable condition for power recovery from process steam. This combination however rarely ever exists. Investigation covering cost estimates for the recovery of solids, liquids, vapors, the heat equivalent of fuel and power are all receiving the attention of progressive manufacturers. This paper, as the title indicates, deals only with the specific problem involved in the investigation of the generation of electric power from process steam.

Wherever steam is employed for process work the boiler pressure is determined by that part of the process requiring the highest temperature, and generally the bulk of the steam is employed at a considerably lower temperature and corresponding pressure. The heat available per pound of available process steam, or so-called *heat drop*, for the generation of power is the difference between the heat content i_1 of the steam at the generating pressure (initial condition) and the heat content i_2 of the steam corresponding to the pressure at which it is employed in the process (final condition), assuming adiabatic expansion (Rankine Cycle). This difference multiplied by the pounds of steam so employed per hour gives the total heat available for power generation by means of reciprocating engines or steam turbines.

The more or less fixed amount of process steam available at the existing process pressure is perhaps more often found to be inadequate to supply all of the power requirements and some means should be adopted to secure as near a balance between the heat available and the heat equivalent of the power requirement as it is possible or practical to obtain with a minimum amount of expense for extra fuel. The heat available (i_1—i_2) per pound of steam may be increased by (1) increasing the initial pressure (2) superheating and (3) decreasing the process pressure (back pressure).

The comparative effect on the basis of per cent gain in the heat available per pound of steam of the fore-

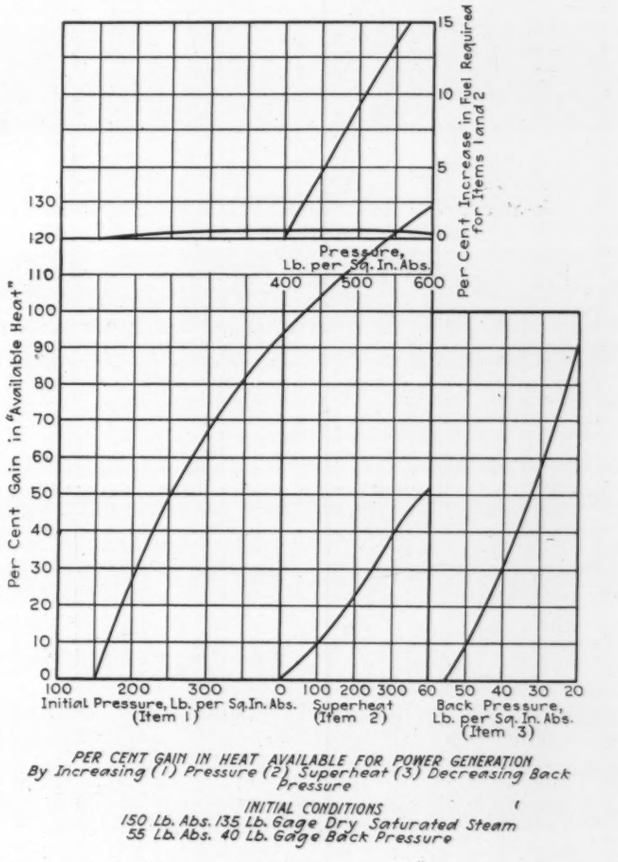

Per Cent Gain in "Available Heat"
Per Cent Increase in Fuel Required for Items 1 and 2
Pressure, Lb. per Sq. In. Abs.
Initial Pressure, Lb. per Sq. In. Abs. (Item 1)
Superheat (Item 2)
Back Pressure, Lb. per Sq. In. Abs. (Item 3)
PER CENT GAIN IN HEAT AVAILABLE FOR POWER GENERATION
By Increasing (1) Pressure (2) Superheat (3) Decreasing Back Pressure
INITIAL CONDITIONS
150 Lb. Abs. 135 Lb. Gage Dry Saturated Steam
55 Lb. Abs. 40 Lb. Gage Back Pressure

FIG. 1. PER CENT GAIN IN HEAT AVAILABLE FOR POWER GENERATION

[1] President, L. A. Harding Construction Co., Buffalo, N. Y.

For Presentation at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

going items as based on an existing condition of 150 lb. per square inch absolute pressure (135 lb. gage) boiler pressure as required for a small part of the process dry saturated steam and 55 lb. per square inch absolute (40 lb. gage) pressure for the bulk of the process steam assumed available for the generation of power, is shown by Fig. 1. The *per cent gain* is evidently the difference between the available heat per pound of steam for the existing condition and the proposed condition divided by the available heat per pound for the existing operating condition.

Suppose, for example, there was found, on comparing the available process steam load and the power load curves and assuming a certain prime mover to be employed, a deficiency of approximately 15 per cent of the steam available for power generation. Fig. 1 indicates that this deficiency can be met in any one of three ways, viz:

1. Increasing the boiler pressure to 175 lb. per square inch absolute or an increase of 25 lb. per square inch.
2. Superheating the steam 150 deg.
3. Reducing the process (back) pressure to 47 lb. per square inch absolute or a reduction in the process pressure of only 8 lb. per square inch.

Increasing the boiler pressure would in all probability require the installation of new boilers, an expensive procedure. Superheaters, if the plant is equipped with water tube boilers, could be installed at nominal expense but would require approximately 7 per cent more fuel for superheating. The reduction in back pressure, if at all possible, would obviously be the natural procedure, as the slight difference in temperature resulting from the reduction in pressure would probably have comparatively little effect on the process. If, however, the deficiency in process steam available for power generation amounted to say 40 per cent, then it is apparent that the boiler pressure must be increased to 225 lb. per square inch absolute or increased a smaller amount and some superheat employed, or use 310 deg. superheat with no increase in pressure, or reduce the process pressure to 36 lb. per square inch absolute (21 lb. gage), a reduction of 17 lb. per square inch. In this case the installation of new boilers designed for the higher pressure and equipped with superheaters would probably be the outcome.

Frequently higher process pressures than necessary are carried and some experimenting to this end may return exceedingly good dividends. The natural procedure in the investigation is to start in the reverse order, and it is frequently found that at least two or possibly all of these gains must be employed to secure the desired result. There remains the possibility of increasing the amount of power that may be generated without additional expenditure for fuel by substituting an electric motor for the small steam engines or steam pumps employed in the plant.

The possibility of effecting boiler plant economies, tending to offset any additional fuel as may be required when items (1) and (2) are considered, should not be overlooked. A study of the existing plant requirements covering both the steam and electric power load curves may result in suggested methods to reduce the peak loads resulting in savings that likewise cost nothing to secure except the time required for the survey and analysis.

The investigation of any specified case generally involves a number of assumptions for the purpose of determining the most economic initial pressure and superheat for the required back pressure as determined by the process. The installation of superheaters for the existing boilers, of additional boiler capacity to be operated at a higher pressure in conjunction with straight non-condensing turbines, mixed pressure, extraction or bleeder type turbines, heat exchangers, steam regenerators or accumulators, and condensing operation utilizing hot well water as a source of hot water supply, etc., are some of the many items which are frequently considered in this connection.

A knowledge of the water rates of prime movers for various conditions of operation is evidently essential to anticipate the amount of power that may be generated from the available process steam. The water rates of high grade reciprocating engines operating non-condensing are somewhat less than the steam turbine of equivalent capacity when operating with the same initial and back pressure. The steam turbine, when operating condensing, however, shows as good an economy as the reciprocating engine.

The turbine offers the advantages that steam may be extracted at various pressures in the expansion, occupies less floor space and, combined with the electric generator, is usually a less expensive combination. There are cases, however, when the most economical machine is essential to give a profitable balance between the heat and power requirements. A comparison of the guaran-

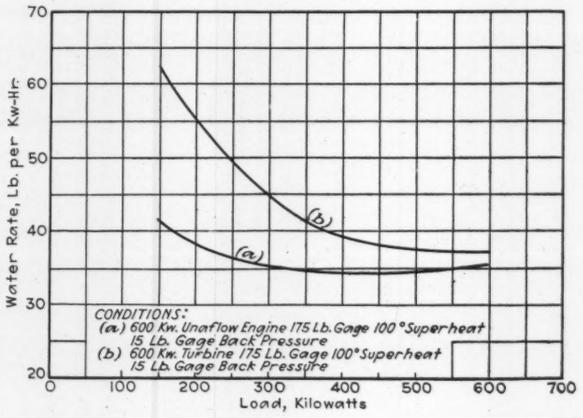
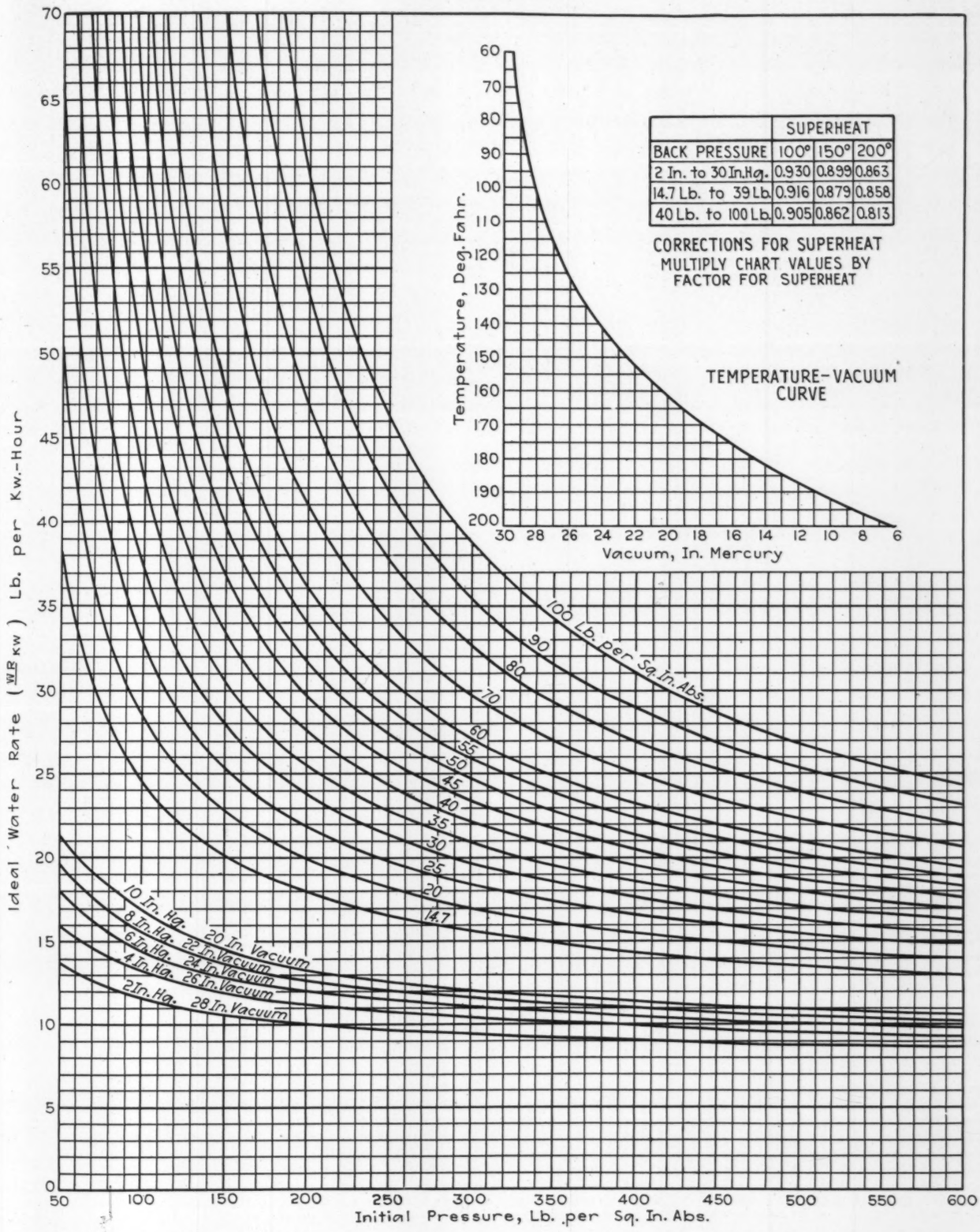


FIG. 2. WATER RATE CURVES FOR NON-CONDENSING OPERATION

teed water rate curves for a 600 k.w. combination of a uniflow engine and generator (*a*) and a steam turbine and generator (*b*), both operating with an initial pressure of 175 lb. per square inch gage at the throttle, 100 deg. superheat and 15 lb. per square inch gage back pressure is shown by Fig. 2.

The rated capacity of generating units adaptable for use in the great majority of manufacturing plants ranges between 200 and 1,500 k.w. The data in this paper refer particularly to the use of steam turbines within these capacities. Steam turbines especially designed to meet exacting requirements naturally are somewhat more expensive and more economical in the use of steam over a limited range of operating loads than a standard commercial design. Standard commercial designs, however, will usually show as good an average economy for fluctuating load conditions, and are ordinarily employed



	SUPERHEAT		
BACK PRESSURE	100°	150°	200°
2 In. to 30 In. Hg.	0.930	0.899	0.863
14.7 Lb. to 39 Lb.	0.916	0.879	0.858
40 Lb. to 100 Lb.	0.905	0.862	0.813

FIG. 3. IDEAL WATER RATE PER KILOWATT HOUR WITH DRY SATURATED STEAM FOR VARIOUS INITIAL PRESSURES AND BACK PRESSURES

when process steam is utilized for power generation.

A steam turbine connected with a generator is rated on the basis of equivalent kilowatt output at full load. The rating of the generator is given in kilovolt-amperes (k.v.a.) with a stated power factor (usually 80 per cent). A 937.5 k.v.a. generator, for example, would require a turbine rating of 937.5 × 0.80 or 750 k.w. The guaranteed water rates for steam turbine generating sets are universally stated in pounds per kilowatt output including the small amount of electric energy required for excitation with alternating current generators.

The theoretical or ideal water rate of a steam engine

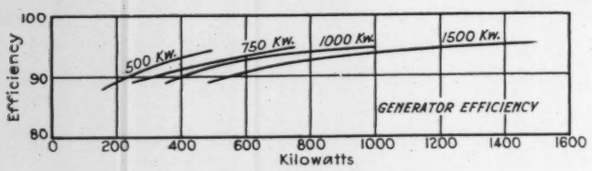


FIG. 4. GENERATOR EFFICIENCIES

or turbine employing steam expansively is determined on the assumption that the expansion takes place adiabatically (constant entropy). The Rankine cycle has for many years been employed as a basis for purposes of comparison. It is assumed, with this cycle, that complete expansion of the steam takes place between the initial and back pressure. The following formula is employed for determining the theoretical or ideal water rate for either steam engines or steam turbines:

i_1 = heat content in B.t.u. per pound of steam at an initial pressure on turbine side of throttle of p_1 lb. per square inch absolute. (The usual pressure drop assumed through throttle valve is 15 lb. per square inch.)

i_2 = heat content in B.t.u. per pound of steam at the back (terminal) pressure p_2 lb. per square inch absolute.

h = heat available for conversion into work in B.t.u. per pound of steam.
$= i_1 - i_2$.

778 = mechanical equivalent of heat (foot-pounds per B.t.u.).

1.34 = ratio of kilowatt to equivalent electrical horse power.

$W\,R_{hp}$ = Ideal water rate — pound per horse power per hour.

$W\,R_{kw}$ = Ideal water rate — pound per kilowatt per hour.
$= 1.34\,W\,R_{hp}$

$$W\,R_{hp} = \frac{33000 \times 60}{778\,h} = \frac{2546}{h} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

ϕ_t = Brake Potential Efficiency ratio of turbine (see table 5).
$$= \text{ratio of } \frac{\text{ideal water rate}}{\text{actual water rate}}$$

ϕ_g = Efficiency of generator (see Fig. 4).

$E\,W\,R_{hp}$ = Expected water rate of turbine-pound per brake horsepower per hour.
$$= \frac{W\,R_{hp}}{\phi_t} = \frac{2546}{\phi_t \times h} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

$E\,W\,R_{kw}$ = Expected water rate-pound per kilowatt hour.
$$= \frac{E\,W\,R_{hp}}{\phi_g} = \frac{2546 \times 1.34}{\phi_t \times \phi_g \times h} \quad \ldots\ldots\ldots\ldots (3)$$

The brake potential efficiency ratio takes into account all losses for the turbine and is a combination of the internal efficiency ϕ_i and the mechanical efficiency ϕ_m or $\phi_t = \phi_i \times \phi_m$. The average mechanical efficiency for the turbines from 300 to 2000 kw capacity may be assumed as 94 per cent to 96 per cent, so that $\phi_i = \frac{\phi_t}{0.94}$ (approx.). The internal efficiency is employed for determining the terminal point for the expansion line of the turbine when drawing this line on a Mollier diagram.

The small and medium size steam turbines employed for the generation of power for industrial plants are of the following types:

Non-Condensing Operation

The back pressure employed varies approximately from 10 to 50 lb. per square inch gage, although there are installations operating with a considerably higher

TABLE 1—HEAT AVAILABLE AND IDEAL WATER RATE LB. PER KW. HOUR DRY, SATURATED STEAM

Back Pressure Lb. Sq. In. Absolute p_2	Initial Pressure Lb. Sq. In. Absolute (p_1)													
	100		150		200		250		300		400		600	
	B.T.U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.
28-in. Vac.	293.0	11.65	318.7	10.7	336.5	10.13	349.6	9.75	359.9	9.48	376.0	9.075	396.0	8.62
26-in. Vac.	257.9	13.2	284.2	12.0	302.2	11.28	315.6	10.8	326.9	10.43	343.2	9.95	364.3	9.37
24-in. Vac.	236.4	14.4	262.7	13.0	281.5	12.1	295.6	11.55	306.6	11.1	323.0	10.57	345.2	9.89
22-in. Vac.	220.4	15.5	246.2	13.85	266.5	12.8	280.6	12.17	291.9	11.7	308.5	11.05	330.8	10.31
20-in. Vac.	208.1	16.4	235.3	14.5	254.5	13.4	268.6	12.7	280.5	12.18	297.5	11.47	320.1	10.66
14.7	140.0	24.4	169.3	20.1	189.5	18.0	205.3	16.6	217.9	15.65	236.5	14.4	261.4	13.06
20	120.0	28.4	149.7	22.8	170.5	20.0	186.6	18.28	199.2	17.12	218.2	15.6	243.3	14.03
25	104.4	32.6	134.7	25.35	156.0	21.9	172.0	19.83	184.9	18.45	204.0	16.71	230.0	14.84
30	91.2	37.4	122.2	27.9	143.5	23.8	159.6	21.4	172.7	19.75	192.5	17.71	219.1	15.58
35	80.4	42.5	111.7	30.54	133.4	25.6	149.4	22.8	162.4	21.0	182.7	18.68	209.3	16.30
40	70.6	48.3	102.0	33.4	123.8	27.6	140.4	24.3	153.6	22.2	174.0	19.6	201.0	16.98
45	62.0	55.0	93.2	36.6	115.2	29.6	132.1	25.8	145.4	23.45	166.0	20.53	193.8	17.60
50	54.0	63.2	85.7	39.8	108.0	31.6	124.6	27.4	138.4	24.63	159.0	21.43	186.8	18.27
55	47.0	72.6	78.7	43.4	100.7	33.9	117.8	28.95	131.6	25.74	152.5	22.4	180.3	18.92
60	40.0	85.3	72.2	47.25	94.5	36.1	111.6	30.55	125.4	27.2	146.5	23.3	174.8	19.50
70	28.4	120.0	60.7	56.25	83.3	41.0	100.6	33.9	114.6	29.8	136	25.1	164.6	20.73
80	17.8	191.5	50.0	68.25	73.0	46.7	90.6	37.6	104.5	32.6	126.4	27.0	155.3	21.96
90	8.4	406.0	41.5	82.2	64.5	52.9	82.0	41.6	96.4	35.4	118	28.9	147.6	23.10
100	.....	.....	33.2	102.8	56.3	60.5	74.1	46.0	88.4	38.6	110.2	30.95	139.8	24.40

TABLE 2—HEAT AVAILABLE AND IDEAL WATER RATE LB. PER KW. HOUR—100 DEG. SUPERHEAT

BACK PRESSURE LB. SQ. IN. ABSOLUTE p_2	INITIAL PRESSURE LB. SQ. IN. ABSOLUTE (p_1)													
	100		150		200		250		300		400		600	
	B.T.U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.
28-in. Vac.	313	10.9	338.7	10.05	358.5	9.51	373.2	9.15	385	8.85	404.0	8.45	431.6	7.9
26-in. Vac.	274	12.45	302.7	11.26	323.0	10.88	338.4	10.06	351	9.72	370.0	9.23	395.6	8.62
24-in. Vac.	251	13.60	280.7	12.14	301.3	11.33	316.4	10.78	329	10.37	348.5	9.79	374.6	9.10
22-in. Vac.	235	14.51	264.7	12.89	285.5	11.95	300.9	11.34	314	10.85	333.5	10.22	359.6	9.48
20-in. Vac.	222	15.38	251.7	13.57	273.0	12.50	288.4	11.81	301	11.32	321.5	10.61	348.6	9.78
14.7	150	22.75	182.7	18.66	205.0	16.65	221.4	15.41	236	14.47	256.5	13.3	288.0	11.85
20	130	26.2	161.7	21.1	185.0	18.45	201.4	16.95	216	15.8	237.5	14.38	268.5	12.7
25	114	29.9	145.7	23.4	169.0	20.2	185.9	18.37	200	17.0	222.5	15.33	252.1	13.50
30	100	34.1	132.7	25.7	156.0	21.85	173.4	19.67	188	18.16	210.0	16.25	240.0	14.2
35	89	38.3	121.7	28.0	145.0	23.5	162.4	21.00	177	19.3	200.0	17.05	229.6	14.85
40	78	43.7	111.7	30.5	135.0	25.2	152.8	22.35	167	20.4	190.5	17.9	221.0	15.41
45	67	50.9	102.2	33.4	126.0	27.1	143.4	23.8	160	21.3	181.5	18.8	215.0	15.87
50	61	56.0	94.2	36.2	118.0	28.9	136.1	25.05	151	22.6	174.0	19.6	205.6	16.60
55	53	64.4	87.2	39.1	111.0	30.7	129.4	26.4	145	23.55	167.1	20.4	200.5	17.0
60	46	74.2	80.7	42.3	105.0	32.5	122.4	27.9	137	24.9	160.5	21.25	192.6	17.70
70	.....	.....	67.0	51.0	93.0	36.7	110.9	30.8	126	27.1	149.5	22.8	184.0	18.55
80	.....	.....	56.7	60.1	82.0	41.6	100.0	34.1	116	29.4	139.0	24.5	175.0	19.5
90	.....	.....	46.7	73.1	72.5	47.0	91.4	37.4	107	31.9	130.5	26.15	164.8	20.7
100	.....	.....	37.7	90.5	64.0	53.4	82.4	41.5	99	34.5	122.0	27.95	158.0	21.6

back pressure. Turbines for this service are of the impulse type usually with only one or two pressure stages, the velocity stages being employed for the first pressure stage. In this case generally only one rotor is used, the steam from the nozzles being redirected through the same row of moving blades as with the re-entry type turbines or two rows of blades attached to the same rotor with a stationary set of blades located between the two rows of moving blades. This is termed a Curtis stage. If more than one pressure stage is employed the succeeding stages are each provided with one velocity stage, i.e., one row of moving blades. Some impulse turbines, however, employ a single rotor for each velocity stage. This arrangement is generally known as a Rateau or multicellular type turbine.

Condensing Operation

Turbines for condensing operation are of the impulse type for the first pressure stage, usually with two velocity stages for the first rotor. The remaining pressure stages are each provided with a single velocity stage. The total number of pressure stages employed varies

TABLE 3—HEAT AVAILABLE AND IDEAL WATER RATE LB. PER KW.HOUR—150 DEG. SUPERHEAT

BACK PRESSURE LB. SQ. IN. ABSOLUTE p_2	INITIAL PRESSURE LB. SQ. IN. ABSOLUTE (p_1)													
	100		150		200		250		300		400		600	
	B.T.U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.
28-in. Vac.	321.0	10.62	349.0	9.78	370.1	9.21	386.0	8.85	398	8.58	417.4	8.18	444	7.7
26-in. Vac.	288.4	12.05	312.5	10.9	334.1	10.2	350.0	9.75	363	9.4	382.4	8.93	410	8.33
24-in. Vac.	260.1	13.1	289.5	11.78	311.6	10.95	328.0	10.4	341	10.0	360.9	9.45	389	8.78
22-in. Vac.	243.4	14.02	273.1	12.5	295.6	11.55	312.0	10.94	325	10.5	345.4	9.9	374	9.14
20-in. Vac.	230.0	14.8	260.2	13.1	282.6	12.06	299.0	11.4	313	10.9	332.9	10.25	362	9.44
14.7	157.4	21.7	189.2	18.02	213.1	16.0	230.5	14.8	245	13.92	266.4	12.81	298	11.47
20	136.0	25.1	168.5	20.22	192.6	17.7	210.0	16.25	225	15.15	247.4	13.8	278	12.3
25	119.4	28.6	152.0	22.42	176.6	19.3	194.0	17.6	209	16.32	231.4	14.75	263	13.0
30	105.6	32.3	138.5	24.62	163.1	20.9	181.0	18.85	196	17.4	218.4	15.61	251	13.6
35	93.4	36.6	126.5	26.95	151.6	22.5	170.0	20.05	185	18.45	207.9	16.43	241	14.18
40	83.0	41.1	117.0	29.1	141.6	24.1	160.0	21.3	175	19.5	198.4	17.2	231	14.8
45	73.4	46.5	107.5	31.7	132.6	25.7	151.0	22.6	166	20.55	189.4	18.04	223	15.3
50	65.4	52.2	99.5	34.3	124.6	27.2	143.0	23.9	159	21.45	182.4	18.8	215	15.9
55	57.0	59.9	91.5	37.3	117.6	29.0	136.0	25.1	153	22.3	174.4	19.58	208	16.41
60	.....	.....	84.5	40.4	110.6	30.9	129.5	26.4	145	23.5	168.4	20.3	202	16.9
70	.....	.....	71.5	47.7	98.1	34.8	117.0	29.2	133	25.7	156.4	21.8	191	17.9
80	.....	.....	60.0	56.9	86.6	39.4	106.0	32.2	122	28.0	145.9	23.4	180	18.97
90	.....	.....	49.5	69.0	77.1	44.3	97.0	35.2	113	30.2	136.9	24.95	171	20.0
100	.....	.....	39.5	86.4	67.6	50.5	88.0	38.8	105	32.5	128.9	26.5	163	20.95

from 6 to 12 for impulse turbines (multicellular) with different manufacturers. The stages, after the first pressure stage, are either of the single velocity stage for each pressure state (Rateau type) or a combination impulse and reaction stage, Parsons type with plurality of stages. The chart (Fig. 3) gives the values of $W\,R_{kw}$ covering various initial and back pressure conditions for steam initially dry and saturated. When the steam is initially superheated the values as read from the chart are to be corrected by the superheat factors given in the accompanying table. The factors are average values for the range of back pressures indicated.

Tables 1, 2, 3 and 4 give values of h and $W\,R_{kw}$ for various pressures and superheat. The values given for h were read from a Mollier diagram by the late G. A. Goodenough. The condition of the steam at various pressure stages in its passage through the turbine is conveniently determined by means of the Mollier Chart, as later shown by several examples. In order to draw the expansion line on this diagram with sufficient accuracy for the purpose at hand, approximate values for pressure stage efficiencies are generally assumed, or the method recommended by E. H. Brown and M. K. Drewry. (*Trans. A. S. M. E.*, 1923) is employed. The internal stage efficiency (ϕ_s) as here employed is the ratio of the heat converted into work by the rotor or rotors for a single pressure stage to the heat given up ("heat drop") by adiabatic expansion of the steam between the initial and terminal pressures for the pressure stage under consideration per pound of steam.

Values of ϕ_s

The following values of ϕ_s may be employed:

Curtis type stage (2 velocity stage to one pressure stage) usually employed in all multicellular turbines for the first stage also in conjunction with reaction turbines (Parsons type). $\phi_s = .55$ to .65 Pressure drop 90 to 120 lb. per square inch. Rateau stages (1 velocity stage per pressure stage) employed in multicellular turbines after the first stage $\phi_s = 0.75$ to 0.85.

The *heat drop* per stage following the Curtis stage is

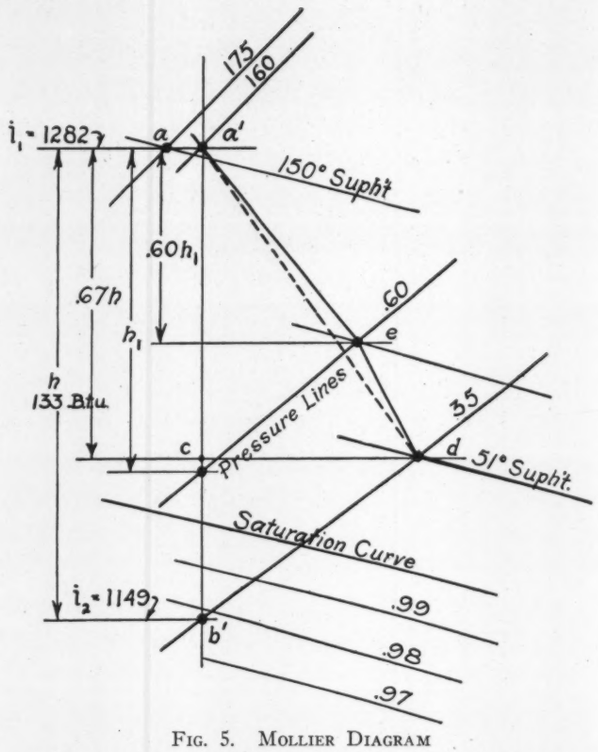


FIG. 5. MOLLIER DIAGRAM

a. State of Steam Entering Throttle Valve, 175 lb. Per Square Inch 150 Deg. Superheat.
a'. State of Steam for 15 lb. Drop in Pressure Through Valve, 160 lb. Per Square Inch Absolute 156 Deg. Superheat.
b. Theoretical State After Expanding from 175 lb. Per Square Inch Absolute to 35 lb. Per Square Inch Absolute.
b'. Theoretical State After Expanding from 160 lb. Absolute to 35 lb. Absolute Pressure.
d. Final Estimated State of Exhaust from Turbine, 35 lb. Absolute 318 Deg. Fahr. or 59 Deg. Superheat.

TABLE 4—HEAT AVAILABLE AND IDEAL WATER RATE LB. PER KW.HOUR—200 DEG. SUPERHEAT

Back Pressure Lb. Sq. In. Absolute p_2	Initial Pressure Lb. Sq. In. Absolute (p_1)													
	100		150		200		250		300		400		600	
	B.T.U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.	B. T. U.	W. R.
28-in. Vac.	331.0	10.3	361.0	9.45	382.0	8.94	398.2	8.56	412.0	8.28	432.0	7.9	459.7	7.43
26-in. Vac.	293.0	11.65	323.4	10.55	344.6	9.9	361.7	9.44	376.0	9.08	396.5	8.60	426.7	8.01
24-in. Vac.	269.0	12.7	299.7	11.38	321.6	10.61	338.7	10.07	354.0	9.64	374.9	9.1	403.7	8.46
22-in. Vac.	252.0	13.52	283.2	12.05	305.6	11.17	322.7	10.58	338.0	10.1	358.9	9.5	389.7	8.78
20-in. Vac.	238.0	14.32	269.7	12.66	292.6	11.66	310.2	11.0	325.0	10.5	346.9	9.84	378.7	9.04
14.7	164.0	20.8	198.0	17.25	221.0	15.43	240.2	14.2	255.0	13.4	279	12.22	310.0	11.0
20	144.0	23.7	176.4	19.35	200.6	17.0	219.2	15.57	233.5	14.6	258.4	13.2	292.5	11.77
25	126.5	27.0	159.7	21.36	184.6	18.5	202.7	16.83	218.5	15.6	243.9	14	275.5	12.37
30	111.8	30.5	146.0	23.4	170.6	20.0	189.7	18.0	206.0	16.55	229.9	14.83	262.2	13.00
35	98.0	34.8	133.7	25.5	159.1	21.4	178.2	19.15	193.8	17.6	218.9	15.6	251.0	13 57
40	88.0	38.8	123.7	27.6	149.1	22.9	168.2	20.3	184.0	18.55	208.9	16.3	243.0	14.05
45	78.0	43.7	113.7	30.0	139.6	24.45	159.2	21.4	175.0	19.5	199.4	17.1	235.0	14.52
50	68.0	50.2	105.0	32.5	131.0	26.0	150.5	22.6	168.0	20.3	192.0	17.78	226.2	15.1
55	59.0	57.9	97.7	34.9	123.6	27.6	143.7	23.75	159.0	21.45	184.9	18.45	219.5	15.5
60	.....	.....	89.7	38.0	116.6	29.3	136.7	25.0	154.3	22.1	178.4	19.1	213.0	16.05
70	.....	.....	76.2	44.8	103.6	32.9	124.2	27.5	140.0	24.4	165.9	20.6	200.5	17.0
80	.....	.....	63.7	53.5	91.6	37.2	112.7	30.3	130.0	26.25	155.4	21.95	192.8	17.7
90	.....	.....	52.7	64.8	81.1	42.0	102.7	33.2	120.0	28.42	145.9	23.4	183.0	18.65
100	.....	.....	42.0	81.2	71.0	48.0	92.7	36.8	111.0	30.7	137	24.9	173.7	19.63

generally assumed constant. There is no direct relation between the overall potential efficiency (ϕ_t) of the turbine and the various stage efficiencies (ϕ_s).

A straight line drawn on the Mollier diagram between the initial and final condition points as a'-d (Fig. 5) is frequently employed as representing the expansion line for approximate determinations of the condition of the steam at various pressure stages. The lower internal efficiency of the Curtis stage, however, generally makes it inadvisable to employ this method.

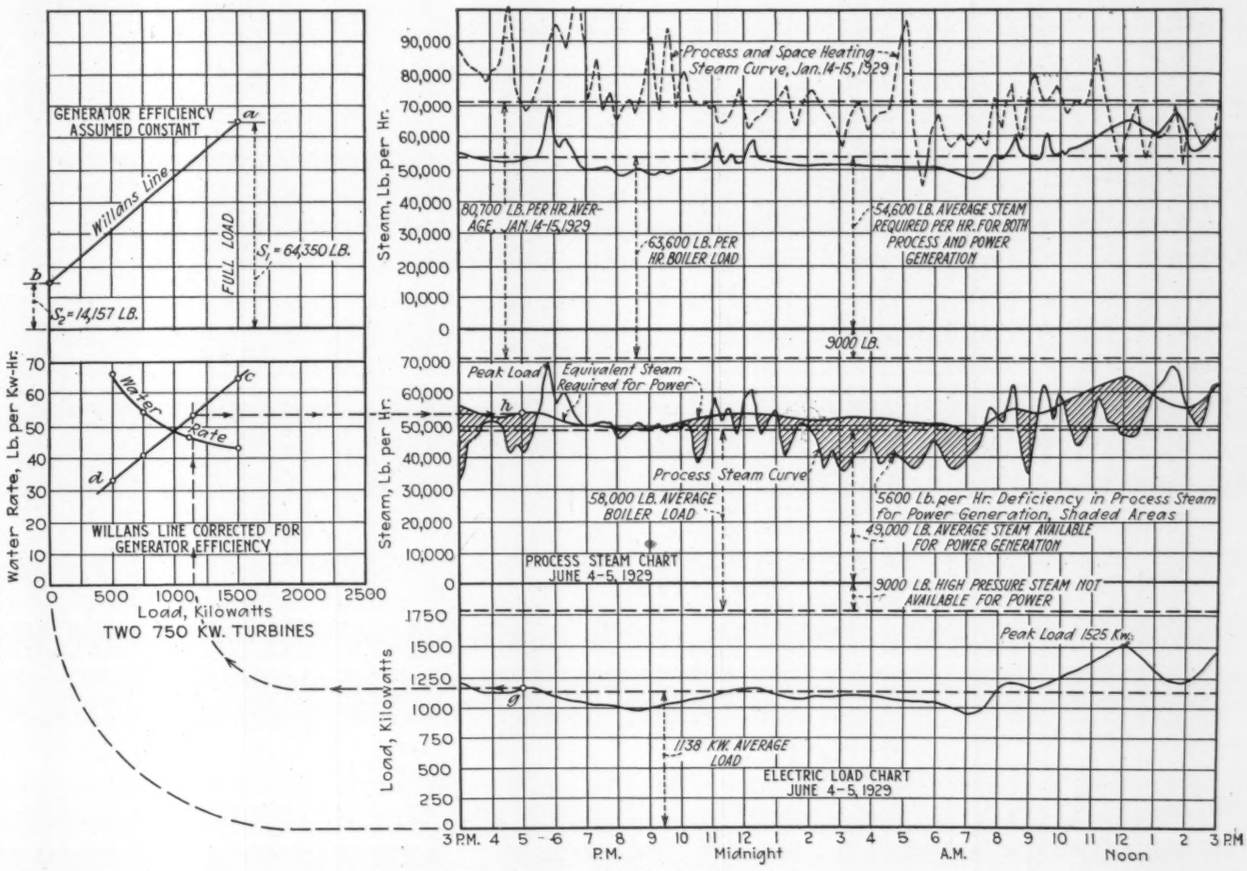

GENERATOR EFFICIENCY ASSUMED CONSTANT
Willans Line
FULL LOAD
S_1 = 64,350 LB.
S_2 = 14,157 LB.
Water Rate, Lb. per Kw-Hr.
WILLANS LINE CORRECTED FOR GENERATOR EFFICIENCY
Load, Kilowatts
TWO 750 KW. TURBINES
Process and Space Heating Steam Curve, Jan. 14-15, 1929
Steam, Lb. per Hr.
80,700 LB. PER HR. AVERAGE, JAN. 14-15, 1929
63,600 LB. PER HR. BOILER LOAD
54,600 LB. AVERAGE STEAM REQUIRED PER HR. FOR BOTH PROCESS AND POWER GENERATION
9000 LB.
Peak Load
Equivalent Steam Required for Power
Process Steam Curve
58,000 LB. AVERAGE BOILER LOAD
5600 Lb. per Hr. Deficiency in Process Steam for Power Generation, Shaded Areas
49,000 LB. AVERAGE STEAM AVAILABLE FOR POWER GENERATION
PROCESS STEAM CHART JUNE 4-5, 1929
9000 LB. HIGH PRESSURE STEAM NOT AVAILABLE FOR POWER
Peak Load 1525 Kw.
Load, Kilowatts
1138 KW. AVERAGE LOAD
ELECTRIC LOAD CHART JUNE 4-5, 1929
3 P.M. 4 5 6 7 8 9 10 11 12 1 2 3 4 5 6 7 8 9 10 11 12 1 2 3 P.M.
P.M. Midnight A.M. Noon

FIG. 6. STEAM AND ELECTRIC LOAD CURVES

Referring to Fig. 5, it is assumed that the steam is supplied to the turbine throttle at a pressure of 175 lb. per square inch absolute, as indicated on the diagram by a. The loss in pressure through the throttle valve is assumed at 15 lb. per square inch, so that the steam enters the first pressure stage of the turbine with the same heat content at 160 lb. per square inch absolute indicated as a' with a temperature of 518 deg. (156 deg. superheat). If the steam expanded adiabatically from 160 lb. per square inch to an absolute pressure of 35 lb. per square inch as point b', the *available heat* (h) is the difference between the total heat for conditions a' and b' or $h = 1282\ (i_1) - 1149\ (i_2) = 133$ B.t.u. per pound of steam. $W\ R_{kw} = \dfrac{2546 \times 1.34}{133} = 25.6$ (compare with value determined by means of the curves and superheat factors for 160 lb. pressure and 150 deg. superheat Fig. 3).

For an assumed turbine brake potential efficiency (ϕ_t) or 0.63 and generator efficiency (ϕ_g) of 0.944 at full load.

$$E\ W\ R_{kw} = \frac{2546 \times 1.34}{0.63 \times 0.944 \times 133} = 43 \text{ lb. per kilowatt hour.}$$

Fig. 5 shows the estimated expansion line a'-e-d for a two pressure three velocity stage machine. It is assumed that the first pressure stage is a Curtis type stage and that ϕ_s for the first stage is 0.64. For an assumed mechanical efficiency of 0.94, $\phi_1 = \dfrac{\phi_t}{0.94}$ or 0.67 for complete turbine.

The Willans Line

Referring to the diagram located at the upper left hand corner of Fig. 6, if the total pounds of steam for any two loads, as for example, the *no load* (b) and *full load* (a) are known for a throttling steam engine or a steam turbine and a straight line is drawn between these points, the total pounds of steam for any intermediate load is correctly given by the ordinate for that load. The diagram is correct when the rating is based on the brake horsepower of a turbine and the indicated horsepower for an engine. The total steam for any intermediate load may also be calculated by means of the equation:

$$S = S_1 \{x\ (1 - y) + y\} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

S = Total steam per hour for the desired load.

S_1 = Total steam per hour for the full load.

y = Ratio of *no load* steam to *full load* brake horsepower steam.[2]

= Approximately 0.22 for non-condensing turbines.

= Approximately 0.14 for condensing turbines.

= Approximately 0.16 for low pressure condensing turbines (atmospheric pressure — 28-in. vacuum).

x = fraction of full load corresponding to S.

It is necessary to correct the Willans line to allow for a decreasing generator efficiency as the load is reduced below the power developed at full load, as will be later shown by an example.

The water rate for any load is determined by dividing the total steam per hour by the corresponding load.

The example (Fig. 6) illustrates a method employed in plotting an equivalent electric power load curve from an available process steam load curve, or an equivalent steam load curve from a power load curve. It is, of course, desirable to have the actual guaranteed water rates from the several manufacturers, whose machines are being considered, covering the proposed conditions of operation. These water rates will ordinarily be found to vary somewhat between the different types of machines proposed. The correct use of the data herein presented, however, will, it is believed, give sufficiently accurate water rates for the purpose intended. The process steam load curve and the purchased electric power load curve shown were plotted from meter readings recorded June 4 and 5, 1929, at the plant under consideration. The load curves for this date were chosen as representing a typical summer day operation when the electric power load gave a high average and maximum peak. The average steam load in this plant does not vary much from day to day during the summer months.

If it is possible to generate the electric power required by the plant from the process steam plus an amount of steam which is less than the difference between the average winter and summer steam loads, during the summer months at a fair saving over the cost of purchased power, it would appear that the extra steam available during the heating season, as required for space heating, will further reduce the cost and show a greater yearly saving.

TABLE 5—APPROXIMATE FULL LOAD EFFICIENCIES MULTI STAGE IMPULSE STEAM TURBINE—GENERATORS—GEARED TYPE

Rated Cap'y. at Full Load kw.	Brake Potential Efficiency Ratio of Turbine $\varnothing_t$ (1)	Generator Efficiency at Full Load $\varnothing_g$ (2)	Combined Efficiency at Full Load $\varnothing_t \times \varnothing_g$ (3)
	NON-CONDENSING TURBINES		
100	0.47	.....	.....
200	0.48	.....	.....
300	0.61	.....	.....
400	0.63	.....	.....
500	0.64	0.940	0.600
600	0.64	0.942	0.603
750	0.65	0.944	0.614
1000	0.67	0.945	0.633
1250	0.67	0.947	0.634
1500	0.67	0.950	0.637
2000	0.68	0.954	0.649
	CONDENSING TURBINES		
500	0.68	0.940	0.639
600	0.69	0.942	0.650
750	0.70	0.944	0.661
1000	0.70	0.945	0.662
1250	0.70	0.947	0.663
1500	0.70	0.950	0.665
2000	0.70	0.954	0.668

1. Ratio $\frac{\text{Ideal Water Rate}}{\text{Actual Water Rate}}$ includes efficiency of gears.

Turbines of the re-entry type are somewhat less efficient than the values given in table for turbines 500 k.w. capacity and above.

2. 60 cycle—3600 r.p.m.

[2] The values of y vary somewhat for various types of turbines depending on the method of governing (throttling or non-throttling) speed, pressure, number of stages, etc.

TABLE 6

Load k.w.	Fractional Load x	Generator Effy. $\varnothing_g$	Ratio— Gen. Eff. Full Load / Gen. Eff. Frac. Load	Steam per Hr. from Willans Line— Lb. S	Corrected Total Steam per Hr. Lb.	Water Rate k.w. Hr. Lb. E W R_{kw}
1500	1.00	0.944	1.000	64350 (S_1)	64350	42.9
1125	0.75	0.931	1.014	51800	52525	46.4
750	0.50	0.909	1.038	39500	41001	54.6
500	0.33⅓	0.890	1.060	31500	33390	66.8

Steam Load Curve

The steam load curve of this plant for January 14 and 15, 1929, is shown by the dashed line at the top of chart, Fig. 6. The boiler equipment at this plant consists of one 450 and four 308 hp. boilers (nominal rating). All boilers are equipped with stokers and forced draft, the stoker equipment being capable of operating the boilers at 200 per cent rating. The boilers are designed for a pressure of 175 lb. per square inch gage and are not equipped with superheaters. High pressure steam for certain operations is required at the average rate of approximately 9000 lb. per hour, and is fairly constant. The remainder of the steam generated, or 49000 lb. average per hour, it is assumed, will satisfactorily supply the remainder of the requirements (cookers, evaporators, drying, water heating, etc.) with a back pressure at the turbines of 20 lb. per square inch gage. In order to supply dry steam at the exhaust outlet of the turbines it is proposed to install superheaters for all of the existing boilers. The cost of 13000 B.t.u. coal in the bunkers is $3.75 per ton. The average *boiler and grate* efficiency of this plant is 68 per cent. From an inspection of the electric power load curve it is evident that the full load turbine rating required should be approximately 1500 kw. to provide for the peak load. Probably the most practical combination, all things considered, would be the installation of three 750 kw. machines, one of which would be a spare unit.

Assuming an initial throttle pressure at the turbine of 160 lb. per square inch gage (175 lbs. absolute), 150 deg. superheat and 15 lb. drop through throttle with 20 lb. per square inch gage (35 lbs. absolute) back pressure, the ideal water rate with 160 lb. absolute pressure is:

$W\,R_{kw} = 29.2 \times 0.879$ (superheat correction) $= 25.5$ lb. per kilowatt hour.

Assuming an efficiency ratio (ϕ_t) for the turbine of 0.63 and a generator efficiency of 0.944, the expected full load water rate for a 750 kw. turbine-generator set for the assumed condition is:

$$E\,W\,R_{kw} = \frac{25.5}{0.63 \times 0.944} = 42.9 \text{ lb. per kilowatt hour.}$$

A guaranteed water rate approximately 10 per cent lower than this figure can be obtained from some manufacturers.

The total estimated steam required at *full load* for two 750 kw. units is:

$$2 \times 750 \times 42.9 = 64350 \text{ lb. per hour.}$$

It is assumed that the ratio *no load* to *full load* steam is $y = 0.22$, the estimated steam at *no load* being equal to $.22 \times 64350$ or 14157 lb. per hour. The Willans line *a-b* may now be constructed as previously indicated and is evidently based on the assumption that the generator efficiency remains constant for all loads. The corrected total steam per hour, line *c-d*, is shown in the diagram directly below the Willans line and was plotted from the calculated data given in Table 6, which in turn is based on equation (4).

The corrected total steam per hour for the various loads shown was obtained by multiplying the values as determined from the Willans line by the ratio of the generator efficiency at full load to the generator efficiency at the fractional load.

Any point, as *g* on the electric power load, may be translated into equivalent steam load by simply projecting over to the intersection with the total steam per hour line and transferring the equivalent power load to the steam chart, as indicated at *h*. Any point on the available steam chart curve may be translated into equivalent electric power by simply reversing this process. The deficiency in the process steam at various times during the day for generating the electric power required is shown by the shaded areas on the Process Steam Chart. Integrating these areas the average deficiency in process steam to generate the power required is found to be 5600 lb. per hour.

The average process steam available is 49000 lb. per hour as determined by integrating the area under the process steam load curve. The average steam generated by the boilers for all purposes is 58000 lb. per hour and the average required, if all power were to be generated at the plant, would be $58000 + 5600 = 63600$ lb. per hour. The hourly quantities of steam as would be required, if all the power was to be generated, is shown by the steam load curve chart at the top of the figure. This chart shows that the peak load on the boiler plant would remain at 78000 lb. per hour, and is in no way affected by the problem of power generation.

The factor of evaporation for 160 lb. gage dry saturated steam and 200 deg. feed water is 1.0592 and for the same pressure condition with 150 deg. initial superheat is 1.148.

The extra fuel required per hour, if all the power is to be generated, is, therefore:

$$\frac{[(63600 \times 1.148) - (58000 \times 1.059)] \times 971.7}{0.68 \times 13000} = 1274 \text{ lb.}$$

or $\dfrac{1274 \times 24}{2000} = 15.28$ tons per day.

The exhaust from the turbine will have a superheat of approximately 59 deg. (Mollier Diagram Fig. 5). Dry steam will therefore be supplied by the turbines to the process.

The deficiency in process steam could be practically eliminated if the average water rate of the turbines employed could be reduced to $\dfrac{49000}{1138}$ or 43.1 lb. per kilowatt hour. The estimated average rate for the assumed conditions is approximately 49 lb. per kilowatt hour, which would require a reduction in the full load water rate of approximately 6 lb. This would require an ideal water rate ($W\,R_{kw}$) of 20.9 lb., which corresponds to an initial pressure of 250 lb. absolute or 235 lb. gage. This increase in pressure would require a new boiler installation, which is not possible in this plant.

The estimated cost of the additional equipment required, if all the power is to be generated at the plant, is as follows:

3—750 kw. steam turbine generating sets	$49,000
3—Foundations for above machines	1,500
4—Superheaters for 308 h.p. boilers	8,500
Piping and electric wiring	15,000
Turbine house	12,000
Estimated cost to change distributing steam mains for the lower pressure to be employed	4,000
Total	$90,000

The daily cost of producing electric power based on the preceding data is estimated as follows:

Fixed charges (15% of $90,000 ÷ 365)	$ 37.01
Additional fuel required, 15.28 tons at $3.75	57.30
Additional labor (stationary engineers)	20.00
Additional supplies	4.00
Total	$118.31

The average electric power load for the day was 1138 kilowatts. The estimated cost per kilowatt hour to produce this amount of power is therefore:

$$\frac{118.31}{24 \times 1138} = \$.00433$$

The cost of purchased power at this plant averages $0.007 per kilowatt hour. This company would apparently save at least 40 per cent of the present power bill, or approximately $22,000.00 per year, if they generated their own power. This saving represents a return of nearly 25 per cent on the estimated investment involved.

The amount of extra fuel, as previously estimated, may be reduced by the employment of extraction or bleeder type turbines, as later more fully described, by passing the extra steam as required to generate the deficiency in power due to the lack of sufficient process steam through more turbine pressure stages to a final lower back pressure.

It is evident that the extra steam required must be wasted so that any reduction in the amount results in a direct saving in fuel.

It is considerably more tedious to plot the equivalent

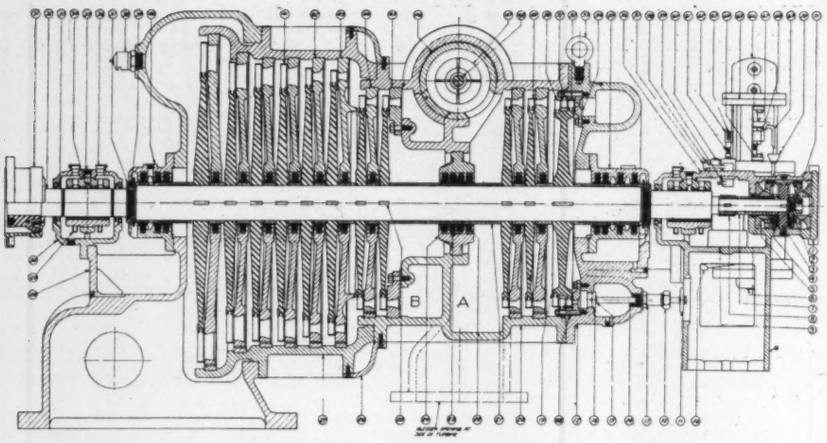


FIG. 7. CROSS SECTION OF MOORE EXTRACTION TURBINE (11 STAGES)

steam chart, when a bleeder turbine is under consideration, to determine the deficiency in process steam. This is due to the fact that an extraction line is required corresponding to each amount of steam it is desired to transfer into electric power or vice versa.

The following method is only a rough approximation, but is frequently employed for preliminary estimates:

Assuming that the extra steam required is expanded in the turbine down to a back pressure of 17 lb. absolute (2 lb. gage) the heat available (Mollier diagram) is 187 B.t.u. per pound. The heat available for 35 lb. absolute back pressure, as previously determined, is 133 B.t.u. per lb. Therefore $1 - \frac{133}{187}$ or 0.25 (approximately) of the excess steam, as previously calculated (5600 lb. per hour), may be saved by the use of bleeder type turbines, or $0.30 \times 5600 = 1680$ lb. per hour (average) and an estimated saving in fuel of 2.5 tons of coal per day. The saving in fuel per year is $2.5 \times 365 \times \$3.75$ or $3439.00. The extra cost of the bleeder turbine equipment is approximately $10,000.00. The fixed charges on this extra cost are $1500.00, so that the estimated net accrued saving is a comparatively small amount ($1,939.00).

Extraction or Bleeder Turbines

(Figs. 7, 8) Steam turbines designed to permit the extraction or bleeding of steam from one or more of the pressure stages are known as extraction or bleeder turbines. The turbine may be designed for either condensing or non-condensing operation. This turbine is principally employed for supplying power in industrial plants.

Practically all manufacturing plants employ some steam for process work and generally always for space heating. The pressure required for the large percentage of steam for these purposes generally does not exceed 40 lb. per square inch gage. The boilers are generally operated at pressures below 225 lb. per square inch when power is to be generated, although in some recent installations 400 lb. pressure is employed. This high pressure is frequently required to effect a heat-power balance. Steam at one or more pressure stages of the turbine is bled to supply the process and the remainder passed through additional stages to a condenser or to the parts of the process requiring the lowest pressure and operating as a non-condensing machine.

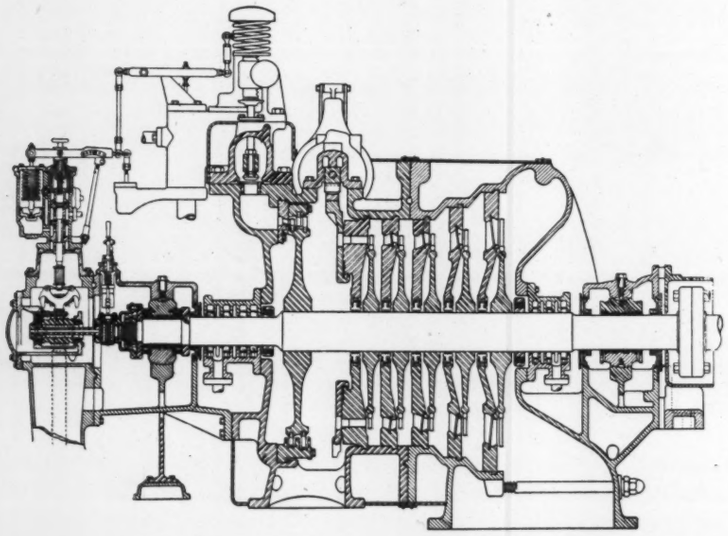
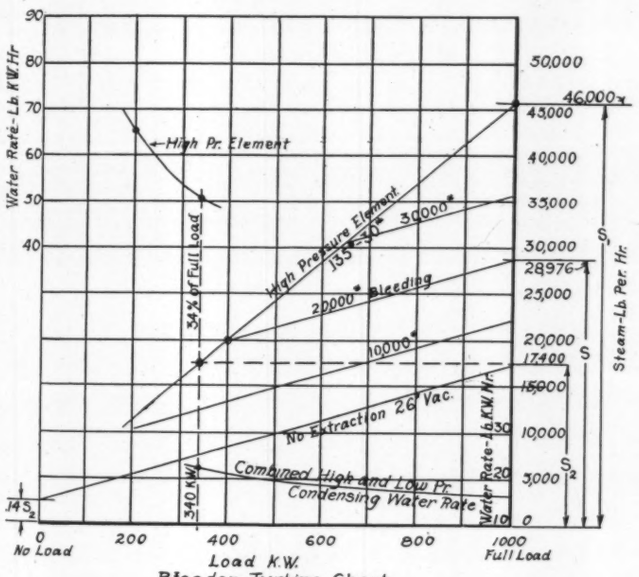


FIG. 8. CROSS SECTION OF GENERAL ELECTRIC COMPANY EXTRACTION TURBINE (6 STAGES)

Condensing operation is often found necessary to secure the amount of power required due to an insufficient supply of process steam at the process pressure or pressures. It is evident, in this case, that more steam must be generated to supply the deficiency, as was shown to be the case in the previous example, and that less extra fuel will be required if the machine is operated condensing.

Bleeder turbines are equipped with an automatic bleeder valve, which admits steam to the low pressure end of the turbine and maintains the bleeder pressure constant. The bleeder valve opens the low pressure steam port or ports with an increasing area as the bleeder pressure rises due to a decreasing demand for process steam, passing the extra steam as may be required for power generation to the low pressure stages of the turbine. An economical application of the bleeder turbine operating condensing is that of supplying hot water for a process which requires both steam and hot water.

FIG. 9. EXTRACTION TURBINE WATER HEATING

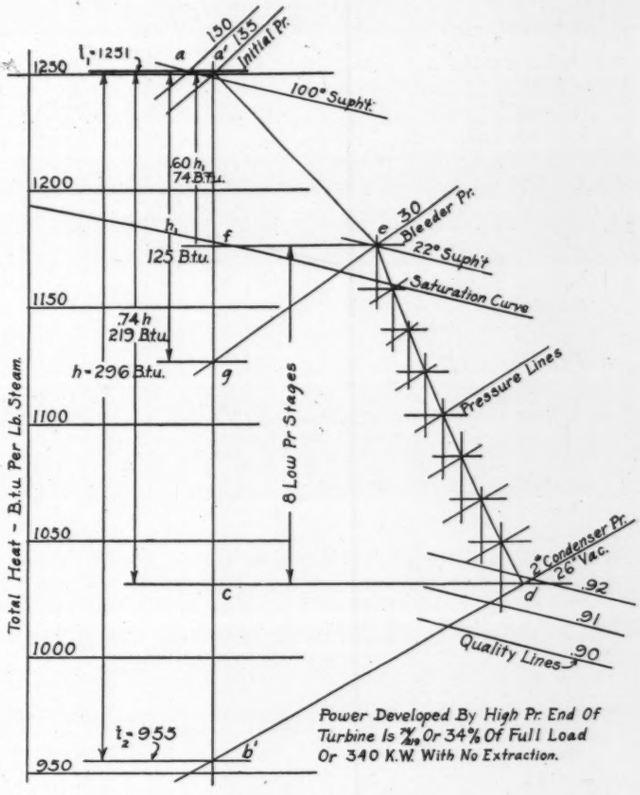


Bleeder Turbine Chart

Conditions.
Throttle Pressure 150 LB. Sq In. Abs. 100° Suph't.
Drop. In Pr. Across Throttle Valve 15 Lb. Sq. In.
Bleeder Pr. 30 Lb. Sq. In. Abs.
Back Pr. 26" Vacuum.

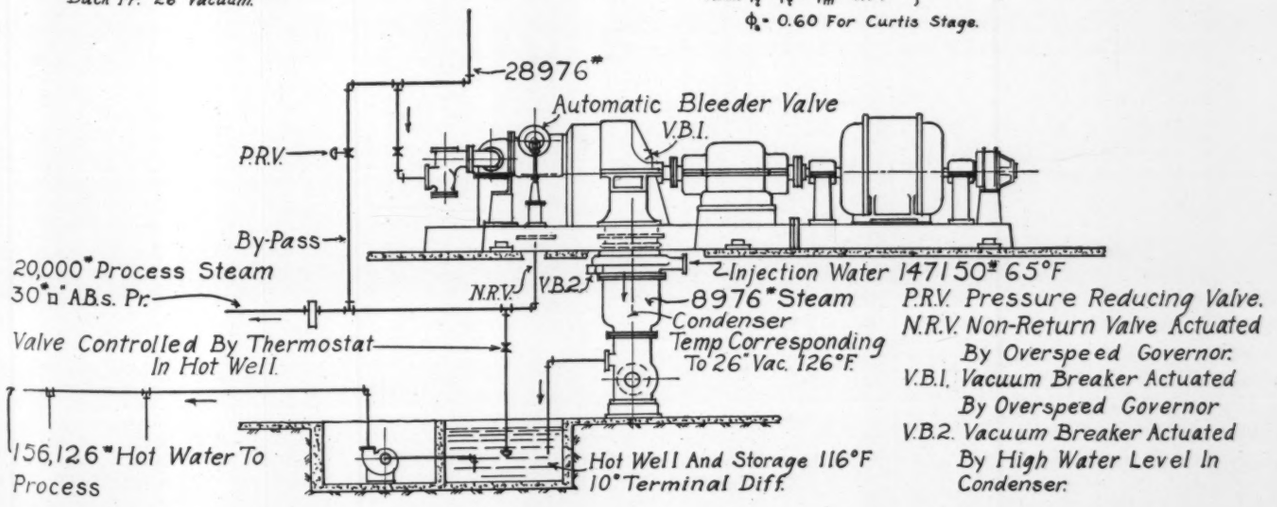
Water Heating for Process Work

When a considerable percentage of the total process steam is required for heating water, the most economical method of obtaining this hot water is from the condenser hot well of a bleeder or extraction type turbine. The bleeder steam is extracted or drawn from the turbine at the pressure stage desired, while the steam required for the water heating is passed on through the low pressure stages of the turbine to the condenser. It is quite evident that more power may be generated in this manner for a given weight of steam employed for water heating


Total Heat - B.t.u. Per Lb. Steam.
150 or 135 Initial Pr.
100° Suph't.
60 h1 74 B.T.u.
30 Bleeder Pr.
22° Suph't
125 Btu.
Saturation Curve
74 h 219 Btu.
h = 296 Btu.
8 Low Pr. Stages
Pressure Lines
2" Condenser Pr. 26" Vac.
.92
.91
.90
Quality Lines
Power Developed By High Pr. End Of Turbine Is 7/20 Or 34% Of Full Load Or 340 K.W. With No Extraction.

Mollier Diagram

ϕ_t = 0.70 Assumed.
ϕ_m = 0.94 - } For Complete Turbine.
Then $\phi_t = \phi_t \times \phi_m$ = 0.74
ϕ_n = 0.60 For Curtis Stage.



due to the considerable difference in water rate between non-condensing and condensing operation. The additional percentage of electric power that may be generated depends on the partial vacuum, the partial vacuum in turn being dependent upon the required temperature of the process hot water.

For all practical purposes the same weight of steam will be required to heat a given weight of water either at the extraction pressure or the condensing pressure. The temperature of the condenser hot well water is assumed the same as the temperature of the hot water employed in the process.

There exists a difference in temperature between the temperature of the steam corresponding to the condenser vacuum and the hot well temperature (*terminal difference*) from 5 deg. to 15 deg., so that the temperature corresponding to the partial vacuum will be approximately 10 deg. higher than the hot well temperature. The absolute pressure corresponding to this temperature is, of course, the terminal pressure for the turbine.

The following example illustrates the employment of an extraction type turbine for this purpose:

Required the electric power that can be generated when the process steam requirement is 20000 lb. per hour (extraction) at 30 lb. per square inch absolute pressure and the hotwater requirement is 147,150 lb. per hour at 116 deg. fahr. The initial temperature of the water is assumed to be 65 deg. fahr. To heat this amount of water per hour requires

$$\frac{147150 \times (116 - 65)}{1000} = 8976 \text{ lb.}$$

steam per hour (latent heat assumed to be 1000 B.t.u. per pound). This is the amount of steam to be passed through the low pressure stages of turbine to the condenser. The total steam available for power generation is $S = 20000 + 8976$ lb. $= 28976$ lb. per hour.

The initial condition of the steam at throttle valve is assumed as 150 lb. per square inch absolute (p_1) 100 deg. superheat as located on the Mollier chart at a (Fig. 9). The pressure drop across the throttle is assumed as 15 lb. per square inch, so that steam enters the first stage of turbine at 135 lb. per square inch absolute with 102 deg. superheat located at a'.

The temperature of the hot water required is 116 deg. fahr. and assuming a 10 deg. *terminal difference* the final steam temperature will be 126 deg. fahr. If any appreciable loss in temperature occurs in the hot water piping between the hot well storage and the point where the water is required for use, the loss may be made up by by-passing the required amount of process steam into the hot well. The temperature (126 deg.) corresponds to an absolute pressure (p_2) of 4-in. mercury of 26-in. vacuum (referred to a 30-in. barometer).

Pass downward on the constant entropy line to the intersection b' with the 4-in. pressure line. The length of the line a'-b' represents the available energy h or 297 B.t.u. per pound of steam for adiabatic expansion between the pressure $p_1 = 135$ lb. square inch and $p_2 = 2$ lb. per square inch. It is only possible to convert a certain percentage ϕ_t of the available energy into work delivered by turbine shaft, which in this example is assumed as 0.70 for the complete turbine. The mechanical efficiency ϕ_m is assumed as 0.94. The internal efficiency ϕ_1 is therefore $\frac{0.70}{0.94}$ or 0.744 for complete turbine.

The heat converted into work by the rotors is then $0.744h$ or 220 B.t.u. as represented by the length of line a-c to scale. The length c-b' to scale represents the heat per pound of steam used to raise the quality of the steam.

A horizontal line through c intersects the pressure line p_2 at d which gives the quality of steam to condenser or 0.917. The stage efficiency of the high pressure end of the turbine will be considerably less than that of the whole machine as previously indicated. A nine stage multicellular turbine is here assumed to be employed with a Curtis type high pressure stage operating between an initial pressure of 135 lb. per square inch absolute and a terminal (bleeder) pressure of 30 lb. per square inch absolute. The available energy h_1 by the expanding steam between the initial condition and 30 lb. absolute bleeder pressure is 125 B.t.u. per pound (a'-g). The high pressure end is assumed to convert 0.60 (stage efficiency) of this amount or $0.60\ h_1 = 75$ B.t.u. per pound into internal work which locates f. A horizontal line through f intersects the 30 lb. pressure line at e representing the state of the bled steam to process which is, in this case, slightly superheated (22 deg. superheat). The line a'-c-d is the estimated expansion line of the turbine as drawn on a Mollier diagram.

A Willans line chart may be constructed for determining the equivalent amount of electric power that may be generated from a bleeder type turbine at various ratings, as follows (Fig. 9):

The water rate for the complete turbine when developing full load (1000 kw.) assuming $\phi_g = 0.945$ and $\phi_m = 0.94$ is: $EWR_{kw} = \frac{2546 \times 1.34}{220 \times 0.94 \times 0.945} = 17.4$ lb. kilowatt per hour. Total steam for full load is therefore 17400 lb. per hour. The no load steam will be approximately $0.14 \times 17400 = 2436$ lb. The total steam line for no extraction is drawn on the chart as indicated.

When the turbine is operating *no extraction* with all the steam going to condenser (17400 lb.) the high pressure end develops $\frac{75}{220}$ or 0.34 of the total load or 340 kw. The water rate for the high pressure end at this load is: $EWR_{kw} = \frac{2546 \times 1.34}{75 \times 0.94 \times 0.945} = 51.2$ lb. per kilowatt hour.

This locates one point on the total steam line for *all extraction*. When developing the full load (1000 kw.) for all extraction the water rate may be assumed as approximately 10 per cent less or 46 lb. per kilowatt hour or 46000 lb. for the full load. The corresponding value of ϕ_s is .67. The "no load" steam for the high pressure end as found by extending the Willans line through the zero ordinate is 2667 lb.

The approximate total steam required to develop full load when 20000 lb. per hour is to be extracted is:

Steam extracted per hour at 30 lb. per square in. $= (W)$ 20000 divided by 46 $= 434.7$ kw.

Steam to condenser per hour at 26 in.

vacuum = (S-W) 8976 divided by 17.4=515.9 kw.

(S) 29836 lb. (A) 950.6 kw.

A line drawn through the 29836 lb. steam per hour point and parallel with the steam line for no extraction is the total steam line for the turbine when operating with 20000 lb. extraction per hour.

The preceding approximation is based on the assumption that the "no load" steam for the high pressure element and the complete turbine is zero in both cases.

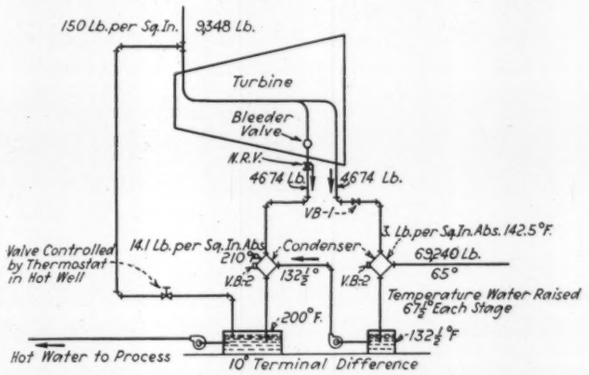


FIG. 10. EXTRACTION TURBINE—TWO STAGE WATER HEATING

The more correct water rates to employ in following formula are as follows:

Water rate for high pressure element (R_1)= S_1—No load steam high pr. element divided by Full load k. w.

Water rate for complete turbine (R_2)= S_2—No load steam complete turbine divided by Full load k. w.

$$\text{or } R_1 = \frac{46000-2667}{1000} = 43.33$$

$$R_2 = \frac{17400-2436}{1000} = 14.96$$

The steam required per hour S to develop full load for W lb. of steam extracted per hour is expressed by the following equation:

$$S = W + \left\{\frac{A-(W-\text{no load hi. pr. element})}{R_1}\right\} \times R_2 \quad \ldots\ldots (5)$$

in which

S = the weight of steam in pounds required per hour to develop full load.

A = full load rating in kw.

W = Weight of steam in pounds extracted per hour.

R_1 = water rate high pressure element.

R_2 = water rate for complete turbine.

$$\text{Then } S = 20000 + \left\{1000 - \frac{20000-2667}{43.33}\right\} \times 14.96 = 28976 \text{ lb. per hour.}$$

The water rate for this condition of operation at full load is therefore:

$$\frac{28976}{1000} = 28.9 \text{ lb. per kilowatt hour.}$$

Any other amount of extraction steam is calculated and the corresponding *total steam per hour* line drawn as above indicated. Extraction lines for 10000, 20000 and 30000 lb. of steam per hour are shown. Owing to approximately 8 per cent (0.917 quality) of moisture in the exhaust for 26 in. vacuum approximately 8 per cent more steam must be passed to the condenser to heat the water required which will also increase, by approximately 8 per cent, the power generated in the low pressure end of the turbine.

Assuming the same amount of steam supplied a straight non-condensing turbine operating at the same initial pressure and superheat with 30 lb. absolute back pressure, the amount of power generated would be approximately: 610 kw. (see Fig. 9).

The increase in power that would be possible to develop by the condenser method, as outlined, over that of employing the process steam at 30 lb. absolute pressure to do the same work is evidently 1000—610 or 390 kw. or a gain of approximately 64 per cent in power without expenditure for additional fuel.

Two Stage Water Heating (Fig. 10)

The previous example illustrates single stage water heating. A more economical method however is to use two or more stages, employing a lower partial vacuum for the first stage and complete the heating by steam bled at a somewhat higher pressure. Assume the same initial pressure as for the previous problem with 15 lb. per square inch drop through throttle and 69240 lb. water per hour to be heated from 65 deg. to 200 deg. Dividing the temperature rise (135 deg.) equally between two stages, a rise of 67.5 deg. for each stage is obtained.

The steam required to heat the water is:

$$\frac{69240 \times 135}{1000} = 9348 \text{ lb. per hour or 4674 lb. per stage.}$$

The temperature of the hot well for the final or second stage is 200 deg. and with a 10 deg. terminal difference gives 210 deg. for the steam temperature to condenser, which corresponds to 14.1 lb. per square inch absolute pressure. The temperature of the hot well for the first stage is 200—67.5 = 132.5 deg. This is also the temperature of the injection water for the second stage condenser.

With a 10 deg. terminal difference the steam temperature for the first stage is 132.5+10 = 142.5 deg., corresponding to a terminal pressure for the turbine of 3.1 lb. per square inch absolute. The theoretical water rate for the complete turbine is 12.4 lb. per kilowatt hour and for the part of the turbine delivering steam at 14.1 lb. per square inch is 19.3 lb. per kilowatt hour. Assuming overall efficiencies of 0.60 and 0.64 the expected water rates become $\frac{19.3}{60} = 32.1$ (2nd stage heating) and $\frac{12.4}{64} = 19.4$ for complete turbine (1st stage heating).

The power that could be generated by two stage heating would be:

Steam bled at

14.1 lb. sq. in. = 4674 divided by 32.1 = 145.6 kw.

Steam to condenser at

3.1 lb. sq. in. = 4674 divided by 19.4 = 240.9 kw.

Total .....9348..... 386.5 kw.

If the single stage heating was employed the power generated would be:

$$\frac{9348}{32.1} = 291.2 \text{ kw.}$$

The gain by two stage heating is therefore:

386.5—291.2 = 95.3 kw. or approximately 33 per cent over the single stage method.

Exhaust Steam Space Heating—Steam versus Hot Water Systems (Fig. 11)

When turbine exhaust steam is to be employed for space heating considerably more electrical power may be generated with a hot water system than with a vacuum steam system on account of the fact that a lower back pressure may be carried at the turbine as will be evident by the following example:

Steam System—Assume a heating load of 100,000 sq. ft. of equivalent direct steam radiation or 24,276,000 B.t.u. per hour or 24,276 lb. of steam per hour (latent heat assumed as 1000 B.t.u. per pound). Initial steam pressure 200 lb. absolute, no superheat and 20 lb. absolute (5 lb. gage) back pressure.

The ideal water rate for the turbine $WR_{kw.}$=20. The expected water rate for an assumed combined efficiency of turbine and generator of 60 per cent is $\frac{20}{60} =$

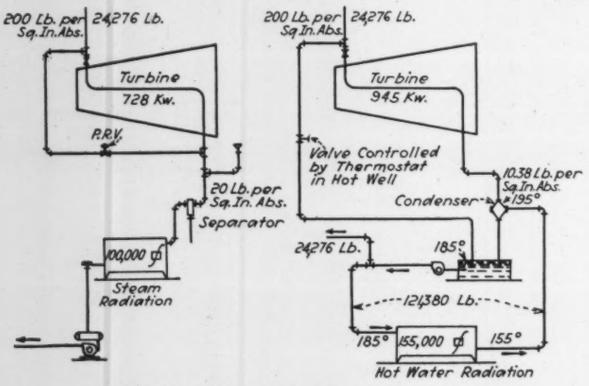


FIG. 11. EXHAUST STEAM HEATING

33.3 lb. per kilowatt hour. The power that could be generated with the vacuum steam system is therefore $\frac{24276}{33.3}$ =728 kw.

Hot Water System—Assuming an initial water temperature of 185 deg. and final temperature of 165 deg. for the radiation, the amount of water to be circulated is:

$$\frac{24{,}276{,}000}{185\text{-}165} = 1{,}213{,}800 \text{ lb. per hour.}$$

The initial temperature of the injection water will be 165 deg. and the final temperature 185 deg. The steam temperature for a 10 deg. terminal difference is 185+10 or 195 deg. corresponding to an absolute pressure of 10.38 lb. per square inch. Assuming a combined efficiency of generator and turbine of 63 per cent the expected water rate ($E\ W\ R_{kw.}$) is $\frac{16.2}{.63}$ or 25.7 lb. per kilowatt hour.

The power that could be generated for the assumed conditions is, $\frac{24276}{25.7} = 945$ kw. The increased power that could be developed is, 945-728 or 217 kw. or a gain of approximately 30 per cent in power with the hot water system. This gain would be somewhat reduced if a sufficiently large vacuum pump were supplied for the steam system to further reduce the back pressure at the turbine.

The diagram on the right of Fig. 12 gives the kilowatts generated per thousand pound of dry saturated steam supplied the turbine at various initial and terminal pressures. When the steam is initially superheated divide the values as determined by the superheat factors (Fig. 3). The expected power per 1000 lb. of steam is determined by multiplying the diagram values (corrected for superheat if any) by the combined efficiency of the turbine and generator (Table 5).

The amount of water which can be heated per 1000 lb. of steam supplied which is also equal to the amount of injection water required at a temperature of 10 deg. less (terminal difference) than the temperature corresponding to the partial vacuum may be read direct from the left chart. See Fig. 3 for temperature—partial vacuum diagram. The heat to be extracted per pound of exhaust steam is here assumed constant at 1000 B.t.u., a fair average value when superheated steam is supplied to the turbine.

The following references will be found useful to those interested in the study of this subject:

1. The Bleeder Turbine—Its History and Theory, by J. L. Moore, Trans. A. S. H. V. E. 1922.

2. Stage Feed Water Heating, by E. H. Brown and M. K. Drewry, Trans. A. S. M. E. 1923.

3. The Value of High Pressures in Industrial Plants, by Wm. F. Ryan, Trans. A. S. M. E. 1925.

4. Exhaust Steam Heating—Mechanical Equipment of Buildings, Vol. I and Steam Turbines, Power Plants and Refrigeration, Vol. II, by Harding and Willard.

5. Balancing Heat and Power in Industrial Plants, by R. V. Kleinschmidt with discussion by various engineers. Trans. A. S. M. E. 1929.

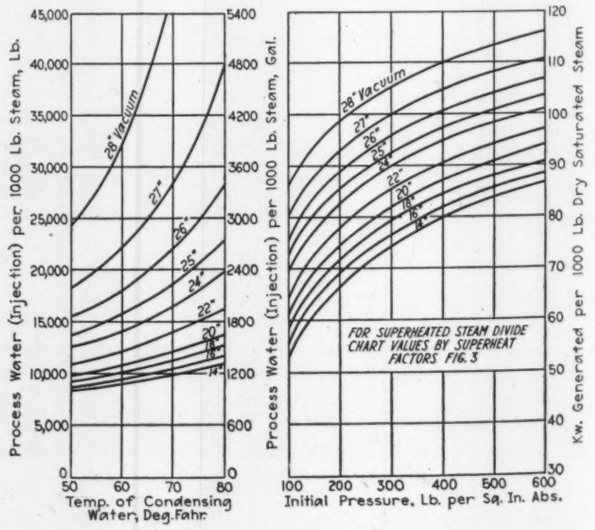


FIG. 12. THEORETICAL K. W. GENERATED AND WEIGHT OF WATER HEATED PER 1,000 LB. OF DRY SATURATED STEAM

6. Heating and Process Loads, by A. R. Acheson, Heating, Piping and Air Conditioning, July, 1929.

7. Steam Turbine Performance, by A. G. Christie, Kent's Mechanical Engineers Handbook.

8. The Application of Low Pressure Turbines, by Francis Hodgkinson, bulletin published by the Westinghouse Machine Co.

9. Mixed Pressure Turbines, by E. D. Dickson, General Electric Review.

10. Steam Turbines, by L. C. Lowenstein, Mark's Mechanical Engineers Pocketbook.

11. Characteristics of Bleeder Turbine, by R. B. Walden. Power 1928.

12. Byproduct Generation in District Heating Plant, by J. H. Walker, Power 1928.

13. Bleeder Turbines, by M. D. Church, Power, June 1929.

14. Multi-Stage Water Heating, by M. D. Church, Power, July, 1929.

15. The Ruths Steam Accumulator by R. A. Langworthy. Trans. A. S. M. E. 1929.

The author is indebted to L. A. Cundall for assistance rendered in calculating Tables 1 to 4 and the Chart Fig. 3.

Report on Commercial Standardization

Acceptances to simplified practice recommendations now in effect, amount to 22,742, of which number 21,488 are from individuals and 1,254 from trade associations, according to an announcement just made public by Ray M. Hudson, assistant director of the National Bureau of Standards, Department of Commerce. His announcement is based on a review of the activities of the Commercial Standardization Group of the Bureau of which he is in charge, for the 3rd quarter of this year.

The number of acceptances to commercial standards now totals 1,758. During the period (July 1 to September 30) for which the review is made, there was a gain of 753 acceptances to simplified practice recommendations and 322 acceptances to commercial standards. The announcement of Mr. Hudson, as it pertains to the various divisions of the commercial standardization group, follows in part:

Two additional simplified practice recommendations, Paper Shipping Tags and Wheelbarrows, received a sufficient number of acceptances from industry during the quarter to warrant their being published by the Division as printed booklets in the Waste Elimination Series. Thirteen other recommendations namely, Vegetable Shortening Containers, Fire Engines, Pyroxylin Coated Fabrics, Granite Curbstone, Industrial Truck Tires, Steel Bathroom and Medicine Cabinets, Hospital Plumbing Fixtures, Tight Cooperage, Dental Hypodermic Needles, Refrigerator Ice Compartments, Soft Fibre Cordage, Colors for School Furniture, and Elastic Shoe Goring, and one regional recommendation on High Volatile Bituminous Coal are now before their respective industries for acceptances. When approved they will be published as part of the Waste Elimination Series, bringing the total number of simplified practice recommendations in effect to 114.

There were five preliminary conferences held by industries during the quarter under the auspices of the Division to consider proposed simplifications. Eleven new proposals were submitted to the Division during the same period. This makes a total of 100 programs now in the development stage.

In July the mimeographed monthly bulletin which the Division has issued for several years was changed to a printed magazine, with a subscription price of $1.00 annually. Its scope was broadened to include the activities of the Commercial Standardization Group. Within 60 days after the magazine had made its debut, the circulation had reached 1,197.

Four commercial standards, Diamond Core Drill Fittings, Template Hardware, Wall Paper and Domestic and Industrial Fuel oils were accepted by their respective industries during the quarter. This makes a total of 12 accepted in less than two years. Five other standards for Standard Weight Malleable Iron or Steel Screwed Unions, Plain and Thread Plug and Ring Gage Blanks, Boys' Blouses, Men's Pajamas, Hickory Golf Shafts, are in process of acceptance.

Two preliminary conferences were held during the quarter to consider proposals for the establishment of additional commercial standards. Seven new projects were received, making the total now on record 41.

Division of Building and Housing

The number of zoned municipalities reported to the Division has materially increased, there now being 795 cities, towns and villages known to have zoning ordinances in effect, as compared to 754 on January 1 of this year.

The latest draft of the amendments to the Mechanics' Lien Act has been compiled and will be presented during October to the National Conference of Commissioners on Uniform State Laws at its meeting in Memphis.

The manuscript of the second volume of the Encyclopedia Series—Standards and Specifications for Nonmetallic Minerals and Manufacturers Thereof—has been submitted for publication. Manuscript is now being prepared for a revised edition of the National Directory of Commodity Specifications.

Mimeographed copies of the "Directory of Government Laboratories" have been sent, in their tentative form, to Federal Government purchasing agencies and others for suggestions as to improvements before the "directory" is placed in its final form.

Manuscripts have been prepared for two informative booklets relating separately to the *certification plan* and the *labeling plan*. The number of Federal specifications covered by lists of Willing-to-Certify Manufacturers is now 267. These lists represent more than 7,800 requests for listing from more than 2,000 firms and concerns.

Referring to foreign interest that has been manifested in the work of the Commercial Standardization Group of the Bureau, Mr. Hudson said that this had been maintained during the quarter. Visitors were received from Belgium, Sweden, England, and Germany, while communications were received from Australia, Canada, England, France, Germany, Italy, Japan, Mexico, New Zealand, Portugal and Switzerland.

NOMINATIONS FOR OFFICERS OF THE SOCIETY FOR 1930

Your Committee appointed to prepare nominations for Officers of the Society for the coming year, 1930, takes pleasure in submitting the following list of nominees:

For President:

L. A. HARDING, 1335 Main St., Buffalo, N. Y.

For First Vice-President:

W. H. CARRIER, Newark, N. J.

For Second Vice-President:

F. B. ROWLEY, University of Minnesota, Minneapolis, Minn.

For Treasurer:

C. W. FARRAR, Buffalo, N. Y.

For Members of the Council:

Three-Year Term

R. H. CARPENTER, New York, N. Y.
JOHN D. CASSELL, Philadelphia, Pa.
ALFRED KELLOGG, Boston, Mass.
JOHN HOWATT, Chicago, Ill.

Respectfully submitted,

NOMINATING COMMITTEE

FREDERICK D. MENSING, Chairman

C. E. BRONSON
R. C. BOLSINGER
H. B. HEDGES
H. M. NOBIS
JAY R. MCCOLL
O. K. DYER
EDWIN C. EVANS, Sec'y.

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—*Section* 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect is own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall if possible announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of THE JOURNAL.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee. In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January 1930:

C. V. HAYNES, Philadelphia
W. T. JONES, Boston
J. F. MCINTIRE, Detroit
F. N. SPELLER, Pittsburgh
A. C. WILLARD, Urbana, Ill.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

2. ELECTION.

a. That the Council shall nominate previous to October 1 of each year, five members to fill vacancies of those retiring at the next Annual Meeting.

b. That the nominations made by Council shall be published in the October JOURNAL of the Society.

c. That prior to December 1 of any year, any ten members, over their own signatures, may nominate one or more additional members for the Research Committee, and such additional nominations shall be placed on the ballot opposite such of the nominations made by the Council as designated by those making the nomination.

d. The election shall otherwise conform to the same regulations as are provided for the election of officers.

e. Vacancies may be filled by the Council, such persons, chosen by the Council, to serve until a successor can be elected at the next annual election.

Air Infiltration through Various Types of Brick Wall Construction[1]

By G. L. Larson[2] (Member), D. W. Nelson[3] (Member), and C. Braatz[4] (Non Member), Madison, Wisconsin

Introduction

SINCE the fall of 1927, a program of cooperative research concerning air infiltration through various types of brick wall construction, sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Wisconsin, has been in progress. The first results of this research appeared as a paper entitled, Effect of Frame Calking and Storm Windows on Infiltration Around and Through Windows, which was presented at the semi-annual meeting of the society in June, 1928. Preliminary to the report contained herein, a second paper entitled Air Infiltration Through Various Types of Brick Wall Construction, was presented at the annual meeting of the society in January, 1929.

This report is in conclusion to the latter paper and contains results obtained on plain, plastered, and painted brick walls.

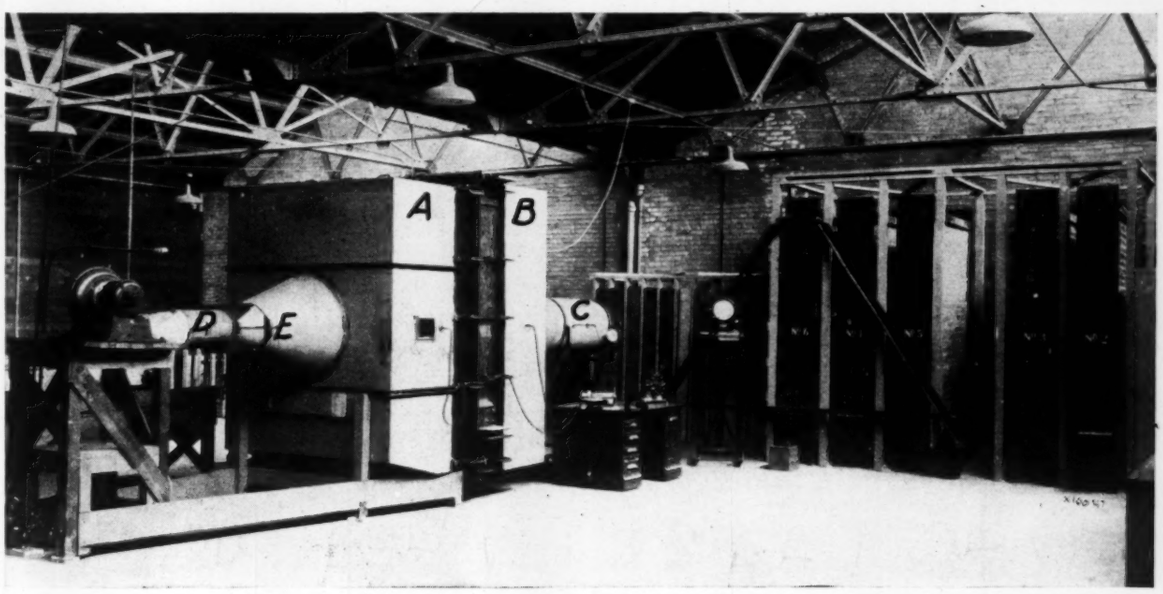


FIG. 1. GENERAL LAYOUT OF TEST EQUIPMENT

Description of Apparatus

The test equipment used to determine air leakage or infiltration is shown in Figs. 1 and 2. Briefly, it consists of the following: The pressure chamber, *A*, and the collecting chamber, *B*, between which the wall is placed. The joints between the steel channel frames, in which the walls are constructed, and the chambers *A* and *B*, were made air tight by clamping a rubber gasket between the flange of each chamber and the corresponding flange of the steel frame, in a manner which is shown in Fig. 2. To facilitate setting up a wall for testing, the collecting chamber side of the machine is mounted on a small four-wheel truck which can be moved back about 10 ft. from the pressure chamber on a narrow gauge track.

Artificial wind pressure is produced by a small motor-driven blower, shown at the extreme left of Fig. 2. The pressure drop through the wall was controlled by means of a damper on the intake of the blower, a relief damper, *D*, and a sliding damper, *E*, located in the connection between the blower and the pressure chamber, at right angles to the flow of air. The pressure drop through the wall, from which the corresponding wind velocity was computed, was measured with an inclined draft gage, *F*.

The amount of air which filtered through a wall was measured by interchangeable orifices mounted at the end of orifice box, *C*, and ranging in size from ⅜ in. to 1¾ in. in diameter, accurately machined in accordance with Durley's[5] specifications. A coefficient of 0.6 was applied to all orifices, in conformity with Durley's results. The accuracy of the small size orifices was checked

[1] Final report on Brick Wall Construction, preliminary report of which was presented at January, 1929, meeting of the Society.

[2] Chairman, Department of Mechanical Engineering, University of Wisconsin.

[3] Assistant Professor of Steam & Gas Engineering, University of Wisconsin.

[4] Instructor in Steam & Gas Engineering, University of Wisconsin.

For presentation at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

[5] On the Measurement of Air Flowing into the Atmosphere Through Circular Orifices in Thin Plates and Under Small Differences of Pressure, by R. J. Durley. Volume 27. *Transactions A. S. M. E.*

by means of a gas meter. A Wahlen gage, *G*, was used to determine the pressure difference existing between the orifice box and the atmosphere, into which the measured air was dischargd.

General Description of Walls

All walls were built into frames constructed of 15 in., 33 lb. steel channels as shown in Fig. 2. The A-shaped frame which appears in the foreground, together with a single roller jack arrangement, attached to the other end of the wall, provides a convenient means of transporting the walls between the test machine and the storage rack, shown in Fig. 1.

Two types of brick were used in the construction of these walls—a hard face brick and a more porous type, commonly known as Chicago clay brick. Water absorption tests were conducted on samples of each type of brick by the National Bureau of Standards, the results of which are tabulated in Table 1. The average dimensions of the hard brick in inches, in the order: Length, width and thickness, were 8⅛, 3¾, 2-11/32, and for the porous type, 8-1/16, 3-9/16, 2-5/16. The average thickness of the vertical and horizontal mortar joints for all of the walls was approximately ¼ and ½ in. respectively.

Three of the walls were slushed with cement-lime mortar; the others with lime mortar. The walls were constructed such as to differentiate between good and poor workmanship. Each of these terms is defined as follows:

Cement-lime mortar: One of cement, one of lime and six of sand by volume, and enough water to make the mixture workable.

Lime mortar: One of lime and three of sand by volume and enough water to make the mixture workable.

Workmanship: Good workmanship is distinguishable from poor workmanship only in the manner of using the mortar. In good workmanship, the spaces between the bricks are completely filled with mortar throughout the thickness of the wall, resulting in a wall which is practically free from voids. In poor workmanship, very little mortar is used between the two outside faces of the wall. The outside appearance of these walls is the same. Fig. 3 is intended to show this difference in workmanship. The poorer wall appears in the foreground.

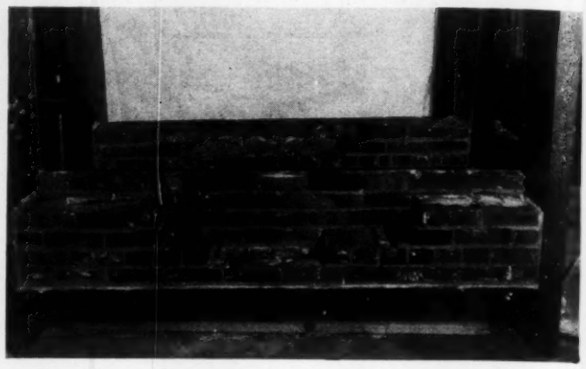


FIG. 3. BRICK WALLS UNDER CONSTRUCTION

TABLE 1—WATER ABSORPTION TESTS OF BRICKS

WATER ABSORPTION (TOTAL IMMERSION) AS PER CENT OF DRY WEIGHT

		1 hr. cold	5 hr. cold	24 hr. cold	48 hr. cold	5 hr. boiling
Red (Hard) Brick.. 10 specimens.....	Ave.	13.1	13.7	14.3	14.8	16.9
	Max.	14.7	15.3	15.9	16.7	18.8
	Min.	10.0	10.3	10.9	11.3	13.4
White(Porous)Brick 10 specimens.....	Ave.	15.0	16.4	17.7	18.2	21.4
	Max.	19.9	21.0	22.0	22.6	24.5
	Min.	11.2	12.8	13.9	14.7	18.3

PENETRABILITY OF BRICK EXPRESSED AS PER CENT OF DRY WEIGHT ABSORBED IN 1 HOUR WHEN IN ⅛-IN. CONTACT WITH WATER

		Per Cent Absorption			Time to Wet Through		
		End	Face	Flat	End	Face	Flat
Red Brick... 10 specimens.	Ave.	4.7	8.8	13.6	Greater than 1 hr..		33 m.
	Max.	5.3	10.1	15.2	Greater than 1 hr..		50 m.
	Min.	3.6	6.1	10.4	Greater than 1 hr..		28 m.
White Brick.. 10 specimens.	Ave.	6.3	9.6	13.7	Greater than 1 hr..		Greater than
	Max.	8.4	13.8	19.1	Greater than 1 hr..		1 hr.
	Min.	3.8	4.8	10.4	Greater than 1 hr..		40 m.

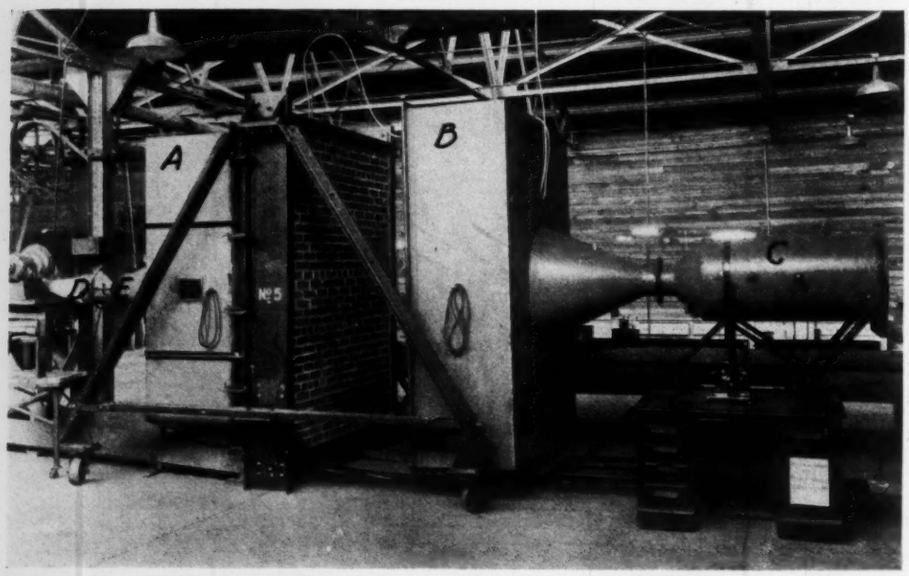


FIG. 2. MACHINE OPEN WITH TEST WALL IN PLACE

The walls were constructed by bricklayers from the Service Department of the University of Wisconsin in a manner comparable to actual building construction practice. Prior to their construction, which took place during the month of February, the bricks were stored out-of-doors. In the opinion of the bricklayers, the brick contained the proper amount of moisture, and the addition of more moisture by soaking was deemed unnecessary.

With the exception of Wall No. 2, the mortar joint between the bricks and the steel channel frame was completely calked with a plastic calking compound, on both sides of all walls. Fig. 4 shows the effect of calking one or both sides of this joint for Wall No. 2. The net area of each wall, excepting No. 7, was 50 sq. ft.

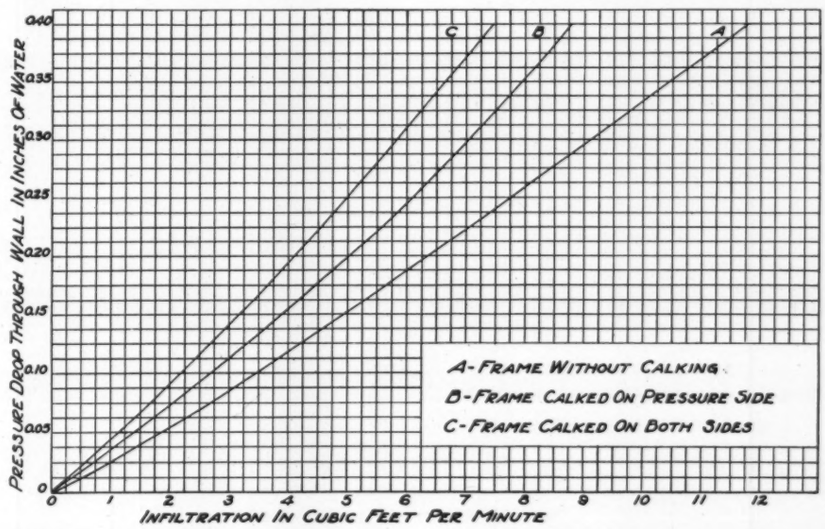


FIG. 4. EFFECT OF CALKING THE MORTAR JOINT BETWEEN THE CHANNEL FRAME AND WALL No. 2

Procedure

Each of the walls was subjected to wind pressures corresponding to wind velocities ranging from about 5 to 30 miles per hour. With the exception of Wall No. 7, which stood for eight months before testing, the original tests of all plain brick walls were made after an aging period of five months. These tests were repeated at the end of two additional months.

Walls No. 4 and No. 5 were later plastered, on metal lath and furring space, for further testing. After removal of the plaster, Walls No. 4 and No. 5 were used for the determination of the effectiveness of several kinds of paint on infiltration.

Wall No. 6 was also re-tested with an application of plaster directly on the brick, without lath.

Wall No. 7 was used only to determine the effectiveness of plaster, no tests being made on the plain brick wall.

A more detailed description of the conditions under which these tests were conducted relative to plaster composition, nature of paint, aging or drying periods, etc., is to be found in the section on Addition of Plaster and Paint to Plain Brick Walls.

Discussion of Results on Plain Brick Walls

Fig. 5 shows the results of the tests on the plain brick walls. The infiltration in cubic feet per hour per square foot of wall surface is plotted against the pressure drop through the wall in inches of water. Each curve was determined from the points of two series of tests; one set of points from the original tests made five months after construction and the other from the check tests made two months later. The difference in values of the original and check tests was found to be less than one per cent for Walls 3, 4 and 6. The check tests on Wall 2 showed 7.1 per cent less infiltration than did the original tests and the check tests on Wall 5 resulted in 2.6 per cent more infiltration than did the original tests. This seems to show that there is no correlation between infiltration and the aging of the walls between the time of the original and check tests. Further, there seems to be no correlation between the humidity at the time of testing and the variation in test results. This comparison is given in Table 2.

TABLE 2—COMPARISON OF HUMIDITY AND TEST RESULTS VARIATIONS

WALL NO.	AV. HUMIDITY ORIGINAL TESTS	AV. HUMIDITY CHECK TESTS	CHANGE IN HUMIDITY EXPRESSED IN PER CENT	VARIATION IN TEST RESULTS PER CENT
2	74.3	40.5	− 45.4	− 7.07
3	55.9	63.4	+ 13.4	− 0.27
4	77.2	58.8	− 23.8	− 0.14
5	65.4	31.7	− 51.6	+ 2.62
6	67.1	36.6	− 45.5	− 0.24

The test results in cubic feet per hour per square foot of wall as shown in Fig. 5 are replotted in Fig. 6 against a uniform scale of velocity in miles per hour instead of against a uniform scale of pressure drop in inches of water. When plotted in this way, the results show how rapidly infiltration increases with increase in wind velocity.

Table 3 gives the infiltration through the various plain brick walls for the range of pressures used in the tests and the corresponding wind velocities ranging from 5 miles per hour to 30 miles per hour. This table also shows a summary of the workmanship, mortar and brick for each of the walls.

The results show that Wall 6 is considerably poorer from the infiltration standpoint than the other walls. Wall 6 was constructed of porous brick and lime mortar applied with poor workmanship. By improving the

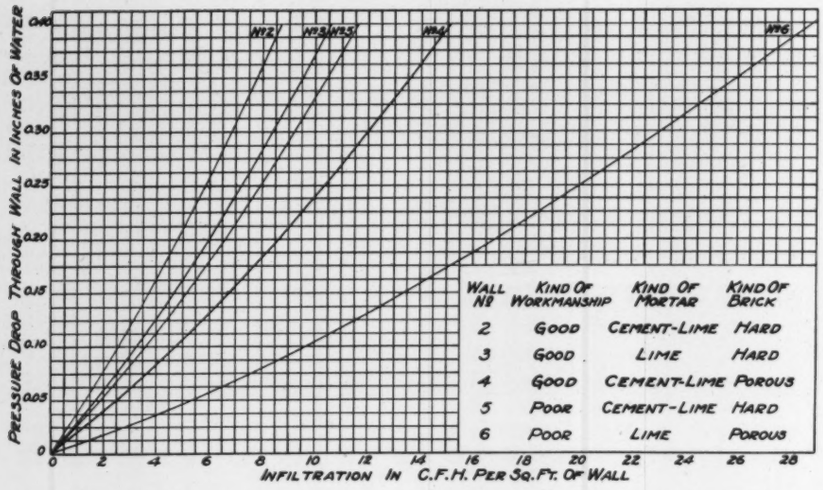

PRESSURE DROP THROUGH WALL IN INCHES OF WATER
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28
INFILTRATION IN C.F.M. PER SQ. FT. OF WALL
NO 2
NO 3
NO 5
NO 4
NO 6
WALL NO | KIND OF WORKMANSHIP | KIND OF MORTAR | KIND OF BRICK
2 | GOOD | CEMENT-LIME | HARD
3 | GOOD | LIME | HARD
4 | GOOD | CEMENT-LIME | POROUS
5 | POOR | CEMENT-LIME | HARD
6 | POOR | LIME | POROUS

FIG. 5. RESULTS OF INFILTRATION TESTS ON THE 13-INCH PLAIN BRICK WALLS

TABLE 3—INFILTRATION IN CU. FT. PER HOUR PER SQ. FT. OF PLAIN WALL

DROP IN PRESSURE INCHES OF WATER	WIND VEL. MILES PER HR.	WALL NO. 2	WALL NO. 3	WALL NO. 4	WALL NO. 5	WALL NO. 6
0.012	5	0.34	0.46	0.71	0.51	1.60
0.048	10	1.30	1.64	2.36	1.83	5.30
0.108	15	2.71	3.45	5.05	3.85	10.35
0.192	20	4.59	5.76	8.31	6.34	16.28
0.300	25	6.85	8.38	12.03	9.22	23.05
0.431	30	9.31	11.30	16.00	12.40	30.80

WALL NO.	KIND OF WORKMANSHIP	KIND OF MORTAR	KIND OF BRICK
2	Good	Cement-lime	Hard
3	Good	Lime	Hard
4	Good	Cement-lime	Porous
5	Poor	Cement-lime	Hard
6	Poor	Lime	Porous

workmanship and using cement-lime mortar rather than lime mortar in Wall 4, the infiltration loss is cut to a little less than 50 per cent. In the case of the hard brick walls, Wall 5, the poorest, allows the passage of 37 per cent as much air as does the poorest wall built of porous brick. The best wall built of hard brick, Wall 2, allows the passage of about 70 per cent as much air as does the poorest wall built of hard brick, Wall 5. The comparison given here between the poorest and best walls for each type of brick is not strictly true, since the poorest of the two hard brick walls had cement-lime mortar as against lime mortar for the poorer porous brick wall.

Had a wall been built of hard brick, lime mortar and poor workmanship, its probable leakage would have been 4.59 cu. ft. per hour per square foot at 15 miles per hour. This figure is arrived at by adding to the leakage through Wall 5 which was built of hard brick, with cement-lime mortar and poor workmanship, the difference between the leakage of lime mortar and cement-lime mortar as applied to hard brick Walls 2 and 3 ($3.85 + 0.74 = 4.59$). The true comparison then between the poorest and best hard brick walls would be that the best hard brick wall would have a leakage value a little less than 60 per cent of that of the poorest hard brick wall. The corresponding comparison for porous brick walls was slightly less than 50 per cent. At 15 miles per hour, the difference between the best and poorest of the hard brick walls would be $4.59 - 2.71 = 1.88$ cu. ft. per hour per square foot of wall, and for the porous brick walls would be $10.35 - 5.05 = 5.30$ cu. ft. per hour per square foot of wall.

This comparison shows that there is a greater variation in infiltration between the best and the poorest walls built of porous brick than between those built of hard brick. This may be due to one of several causes. One possible cause is the variation due to chance. Were a similar set of walls to be built, the results likely would not check exactly the results of this series of tests because of a variation in materials and workmanship. Also, there is a possible cause in the psychology of good materials. A workman instructed to do equally poor work on two walls, one built of hard brick and the other of porous brick, might unconsciously do better work with the better material, the hard brick. It also seems logical to believe that two walls of good workmanship are more likely to be on a comparable basis from a workmanship standpoint than would two walls of poor workmanship, inasmuch as a completely slushed wall is more easily duplicated.

Since poor workmanship consists mainly in leaving voids between the bricks in the interior of the wall, and since the porous brick is likely to be much less uniform in density, it may be that poor workmanship allows passageways for air through short distances from the face of the porous brick to the voids, and then out on the other face of the wall through a short distance of brick. This explanation would require that the hard brick wall passes most of the total infiltration through the mortar joints.

Another probable cause for this greater variation between the best and poorest of porous brick walls as compared to the best and poorest of hard brick walls is in the effect of the porosity of the brick on the proper setting of the mortar. It is likely that the porous brick draws the water from the mortar before it has time to set and consequently causes an opening of pores and a shrinking away of the mortar from the brick surfaces. The bricks, both hard and porous, were considered by the mason to contain the proper amount of moisture for the proper laying up in the walls.

Table 1 shows the water absorption characteristics of the two kinds of brick used. It seems that the absorption test giving the best indication of the drying out effect of the bricks on the mortar would be the total im-

mersion test in cold water for 24 hours, since the bricks are about five-sixths immersed in the mortar and 24 hours is near to the time required for proper setting of the mortar. For total immersion for 24 hours in cold water, a comparison of the figures in Table 1 shows that the hard bricks absorb 81 per cent as much water as the porous bricks. In the absorption tests, the bricks are dry to start with; in building the test walls, the bricks come in contact with mortar when they are somewhere between the dry and saturated condition. Also one face is exposed to the air. Under these altered conditions, it is not known whether the ratio of absorption for the two types of bricks would remain the same as under the absorption test conditions. The tests on air infiltration seem to indicate a greater difference in leakage through mortar applied to hard and porous bricks than the absorption tests would indicate.

The hard brick wall, with lime mortar and poor workmanship synthesized from Wall 5 and a comparison of Walls 2 and 3 would have a probable leakage of 4.59 cu. ft. per hour per square foot at 15 miles per hour. This would be a wall built to the same specifications as Wall 6, except for the difference in brick. Wall 6 had a leakage of 10.35 cu. ft. per hour per square foot at 15 miles per hour. The poorest hard brick wall would then have a leakage of 44 per cent as great as that through the poorest porous brick wall.

The substitution of cement-lime mortar for lime mortar in this poorest hard brick wall would reduce the leakage by 0.74 cu. ft. per hour per square foot of wall at 15 miles per hour, or a saving of 16 per cent. The difference in leakage of Walls 5 and 2 gives the comparison of good and poor workmanship for hard brick walls. The saving in using the better mortar is 1.14 cu. ft. per hour per square foot, or a saving of about 25 per cent. The total reduction in infiltration by using the cement-lime mortar applied with good workmanship in place of lime mortar applied with poor workmanship is 1.88 cu. ft. per hour per square foot for hard brick walls, or a reduction of 41 per cent.

The same comparison of mortar and workmanship cannot be made separately for porous brick walls, since only two walls were tested. However, the best porous brick wall on which cement-lime mortar was applied with good workmanship has a leakage of 5.30 cu. ft. per hour per square foot less than does the poorest porous brick wall, which uses lime mortar applied with poor workmanship, or the reduction is 51 per cent. For hard brick, this reduction was 41 per cent. This seems to indicate that it is more important to use the best mortar and workmanship on porous brick walls than on hard brick walls. This is indicated both by the greater saving in cubic feet of infiltration and by the percentage saving. In actual construction, the item of costs would also enter in choosing material and workmanship. The hard brick is more expensive and the added cost of cement-lime mortar over lime mortar, and good workmanship over poor, would result in a smaller percentage increase in the wall cost than in the case of porous brick walls.

A hard brick wall with lime mortar applied with poor workmanship, it was found, would have a leakage of 4.59 cu. ft. The porous brick wall with cement-lime mortar and good workmanship has a leakage of 5.05 cu. ft. Then the poorest hard brick wall has a leakage 91 per cent as large as that of the best porous brick wall.

Addition of Plaster and Paint to Plain Brick Walls

With the completion of the original and check tests on the plain brick walls, the results of which are shown in Table 3, plaster and paint were applied to some of the walls. The walls used for these additional tests were numbers 4, 5 and 6, and were selected because of their relatively high leakage as plain walls. The same wall was used for both plastering and painting in the case of Walls 4 and 5. The work was done by plasterers and painters from the Service Department of the University of Wisconsin.

The first step taken with Walls 4 and 5 was the application of metal lath and plaster with a furring space. The thickness of the plaster was ¾ in. and the furring space was about the same. The plaster used was a gypsum plaster applied in three coats. The scratch coat consisted of two parts of sand to one part of plaster with hair. The brown coat was proportioned two and one-half parts of sand to one part of plaster with hair. The sand finish coat was made of equal parts of sand and plaster without hair. On Wall 6, three coats of gypsum plaster were applied directly to the brick. The total thickness of the plaster and the composition of the coats were the same as for Walls 4 and 5, except that the brown coat was proportioned three parts of sand to one part of plaster with hair. The walls were allowed to stand three weeks or longer after plastering before testing. The joint between the steel frame and the plaster was sealed with roofing cement.

After the tests of Walls 4 and 5 with the plaster ap-

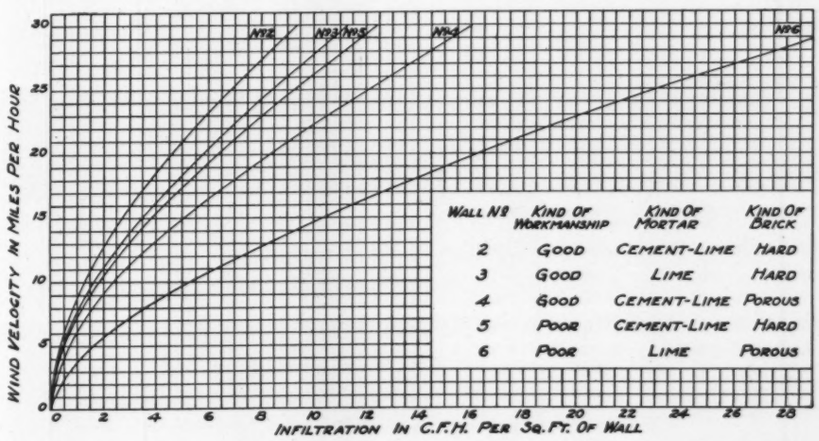


FIG. 6. CHART SHOWING HOW INFILTRATION INCREASES WITH INCREASED WIND VELOCITY THROUGH 13-INCH PLAIN BRICK WALLS

plied were completed, the walls were stripped of the furring, lath and plaster, so that they were in their original plain condition except for the nail holes remaining in the mortar joints from the nailing of furring strips. The walls were re-tested for air infiltration and paint was then applied. All painting was done with a brush.

Three coats of exterior white lead and oil paint were applied to Wall 4 directly on the bricks. This paint was mixed in the proportions of 100 lb. of white lead paste to 4 gal. of linseed oil and ½ gal. of turpentine. The wall was tested after the application of each coat. The first and second coats were applied with just ordinary care with no attempt to fill all crevices. The wall appeared well painted after the application of the first two coats. The third coat was put on with special effort to seal all pores and crevices. This coat was applied with much more than ordinary care. A drying period of one week was allowed after each coat before testing.

One heavy coat of cold water paint was applied directly to the bricks of Wall 5 upon removal of the furring strips, metal lath and plaster. The wall was tested after a drying period of three weeks. After the wall was tested, this coat was washed off as completely as possible with a stiff brush and water; first by hard scrubbing to loosen the paint, and then by flushing off with water. An effort was made to remove it from the mortar joints. The wall was then tested again after standing one month.

Wall 5 was then given two coats of a prepared linseed oil paint containing barium sulphate and zinc sulphide mixed with turpentine for interior use. A test was made after the application of each coat. The first coat dried for two and one-half weeks before testing; the second, five days.

No tests were made of paint applied to plaster.

The results of the plaster and paint tests are shown in Table 4 for Wall 4, Table 5 for Wall 5 and Table 6 for Wall 6. The results are shown graphically in Fig. 7 for Wall 4, Fig. 8 for Wall 5, and in Fig. 9 for Wall 6. The plain wall results before plaster was applied and after plaster was removed have been included in these tables and on the curve sheets for comparison. The one exception to this was in the case of Wall 4, where the plain wall results before plastering were left out, since the curve was almost identical to *C*, the curve for one coat of white lead and oil.

The results show that in each case plastering reduced the infiltration to a very low value. At 15 miles per hour, plastering reduced the leakage through Wall 4 by 94 per cent, through Wall 5 by 96 per cent, and through Wall 6 by 98 per cent. It is to be expected that plaster applied directly to the brick would be more effective in preventing infiltration than plaster applied on lath with a furring space, since the furring space acts as a distributing chamber for air coming through passageways in the brick and mortar. Plaster applied directly, it seems, would tend to close the passageways in the brick and mortar. The percentage figures given above seem to indicate that this is true, since the percentage reduction was the greatest (98 per cent) for Wall 6 which had plaster applied directly. Not too much reliance can be placed on the

TABLE 4—RESULTS OF TESTS ON WALL 4

Wind Vel. Mi./ Hr.	Drop in Pressure Inches of Water	Plain Wall	Furring Metal Lath Plaster	Plaster Removed	1 Coat White Lead and Oil	2 Coats White Lead and Oil	3 Coats White Lead and Oil
5	0.012	0.71	0.05	0.65	0.61	0.61	0.52
10	0.048	2.36	0.14	2.52	2.38	2.31	1.87
15	0.108	5.05	0.28	5.22	4.97	4.76	3.75
20	0.192	8.31	0.47	8.64	8.31	7.78	6.08
25	0.300	12.03	0.68	12.55	12.07	11.27	8.69
30	0.431	16.00	0.92	16.87	16.03	15.18	11.62

TABLE 5—RESULTS OF TESTS ON WALL 5

Wind Vel. Mi./ Hr.	Drop in Pressure Inches of Water	Plain Before Plastering	Furring Metal Lath and Plaster	Plain After Plaster was Removed	1 Coat Cold Water Paint	Cold Water Paint Washed Off	1 Coat White Oil Paint	2 Coats White Oil Paint
5	0.012	0.51	0.02	0.65	0.27	0.27	0.27	0.27
10	0.048	1.83	0.07	2.30	1.03	1.03	1.03	1.03
15	0.108	3.85	0.15	4.61	2.22	2.17	2.13	2.03
20	0.192	6.34	0.25	7.45	3.77	3.37	3.27	3.08
25	0.300	9.22	0.34	10.87	5.58	4.54	4.40	4.12
30	0.431	12.40	0.42	14.86	7.67	5.77	5.61	5.19

difference in these percentage reductions because of the small quantity of air being measured in the tests of plastered walls. For example, at 15 miles per hour for Wall 5, a reduction in leakage of 0.03 cu. ft. per hour per square foot of wall would change the percentage reduction by one per cent. It would seem that the most difficult condition in which to secure a large reduction in leakage would be by plastering on the wall having the

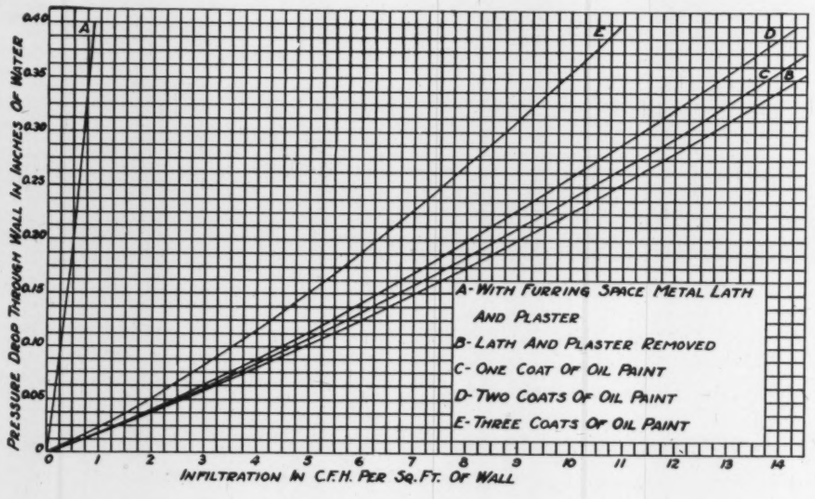

PRESSURE DROP THROUGH WALL IN INCHES OF WATER
INFILTRATION IN C.F.H. PER SQ. FT. OF WALL
A
B
C
D
E
A- WITH FURRING SPACE METAL LATH AND PLASTER
B- LATH AND PLASTER REMOVED
C- ONE COAT OF OIL PAINT
D- TWO COATS OF OIL PAINT
E- THREE COATS OF OIL PAINT

FIG. 7. INFILTRATION THROUGH WALL NO. 4 UNDER VARIOUS CONDITIONS

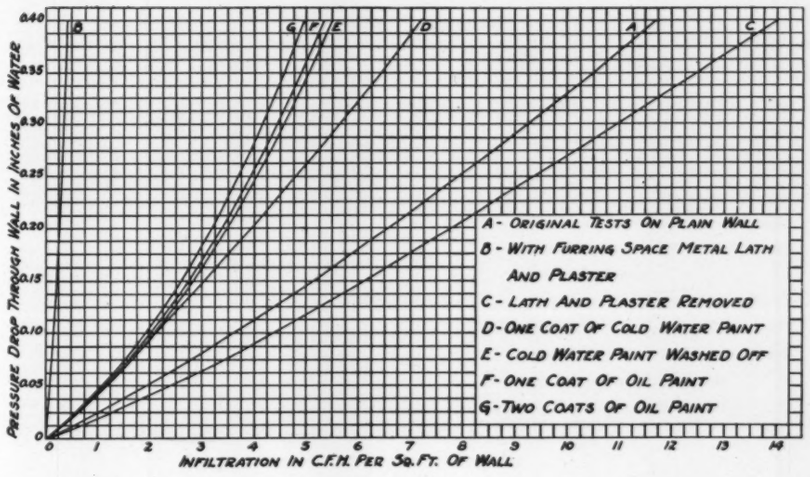

PRESSURE DROP THROUGH WALL IN INCHES OF WATER
INFILTRATION IN C.F.M. PER SQ. FT. OF WALL
A - ORIGINAL TESTS ON PLAIN WALL
B - WITH FURRING SPACE METAL LATH AND PLASTER
C - LATH AND PLASTER REMOVED
D - ONE COAT OF COLD WATER PAINT
E - COLD WATER PAINT WASHED OFF
F - ONE COAT OF OIL PAINT
G - TWO COATS OF OIL PAINT

FIG. 8. INFILTRATION THROUGH WALL NO. 5 UNDER VARIOUS CONDITIONS

TABLE 6—RESULTS OF TESTS ON WALL 6

WIND VEL. MI./HR.	DROP IN PRESSURE INCHES OF WATER	PLAIN WALL	PLASTERED DIRECTLY ON BRICK
5	0.012	1.60	0.03
10	0.048	5.30	0.09
15	0.108	10.35	0.18
20	0.192	16.28	0.29
25	0.300	23.05	0.40
30	0.431	30.80	0.52

greatest leakage as a plain wall. Wall 6 had considerably the highest leakage as a plain wall, which seems to indicate that the greater per cent reduction for plaster applied directly to the brick is real. In service, however, a furred wall would probably have less tendency to crack from any settling occurring in the brick wall and would, therefore, better maintain its resistance to air leakage.

The application of one coat of interior cold water paint reduced the leakage at 15 miles per hour of Wall 5 by 52 per cent. The wall face was completely covered by this coat and no second coat was applied. The reduction in leakage on painting with cold water paint is considered to be due to the clogging of pores and crevices in the brick and more especially in the mortar joints by the paint which was applied as a thick paste. The surface appearance was dull and chalky and gave no indication of effecting a considerable reduction in leakage. Upon testing after the removal of all cold water paint from the surface by scrubbing with a brush and water, the leakage was found to decrease slightly over the leakage secured when the cold water paint was applied. This further bears out the belief that the leakage was reduced by filling openings below the surface. The scrubbing action with water apparently carried additional paint material into pores and crevices.

Wall 5, then in the condition of a plain wall with cold water paint washed off, was treated to two coats of a white oil paint. It would seem that the cold water paint, although filling the crevices, would still be porous and that the oil from the oil paint would bind this porous material together and further reduce the leakage considerably. The first coat gave a reduction in leakage at 15 miles per hour of only 2 per cent and the second coat an additional 4.5 per cent, or a total reduction of 6.5 per cent. The application of two coats to the hard brick seemed to give a smooth and finished surface. The surface was less perfect on the mortar and it would seem that the oil paint was incapable of stopping the leakage through pores and crevices that the cold water paint had not already filled and stopped completely.

White lead and oil paint applied to the bare brick surface of Wall 4 after the furring strips, metal lath and plaster were removed gave a reduction measured at 15 miles per hour of 5 per cent for one coat and 9 per cent for two coats. The brick of this wall was a porous brick and it was observable that two coats did not cover all the imperfections in the brick, although the appearance of the surface was excellent from a distance. An additional coat applied with extreme care in an attempt to cover all imperfections resulted in an additional 19 per cent reduction or a total reduction of 28 per cent for the three coats.

The nail holes made by the nails holding the furring strips in place increased the leakage considerably more on the poor workmanship wall, No. 5, than on the good workmanship wall, No. 4. The test of Wall 4 after the furring strips, metal lath and plaster were removed showed an increase in leakage over that in the original tests attributable to the nail holes of 3.5 per cent measured at 15 miles per hour. The corresponding increase for Wall 5 was 20 per cent. The type of mortar was the same for the two walls. In the case of Wall 4, the joints were completely slushed, whereas in the case of Wall 5, they were not and the nail holes penetrated into the void space in the interior of the wall.

Importance of Sealing Plaster

The tests show that plaster is very effective in reducing leakage to a negligible value. The condition of the plastered walls tested was most favorable to low leakage. No cracks existed in the plaster, and the perimeter against the steel frame channel members was completely sealed. In ordinary building construction, it is doubtful if the full efficiency of the plaster in stopping leakage is ever realized. There is opportunity in the case of a furred wall for the air to travel in the furring space to edges of the plaster sheet and enter the room at the edge of the wood trim such as baseboards, and also to get into the floor construction and enter the room through the flooring joints or through openings made in it such as are made for heating pipes. To obtain the full efficiency of the plaster in stopping air leakage, the plaster sheet should be sealed at all edges. At the baseboard, this would mean the running of the plaster coats down

to the rough floor and also filling the space between the plaster sheet and the brick wall surface with plaster. Any cracks or imperfections due to uneven settling of the building would tend to destroy the effectiveness of the plaster in stopping air infiltration.

To study the effect of imperfect sealing at the baseboard, Wall 7 was plastered and equipped with an 8-in. baseboard. The wall was built of porous brick, lime mortar and poor workmanship and was equipped with a window opening. Gypsum plaster in ¾-in. thickness was applied on metal lath in three coats, and a furring space of about the same thickness was provided. The plaster coats were stopped irregularly within an inch or two of the bottom channel, which corresponds to the floor level in building construction. The baseboard was then applied in the usual manner, except it was screwed to the grounds instead of nailed. The wall was allowed to dry one week before testing. Fig. 10 shows Wall 7 after plastering.

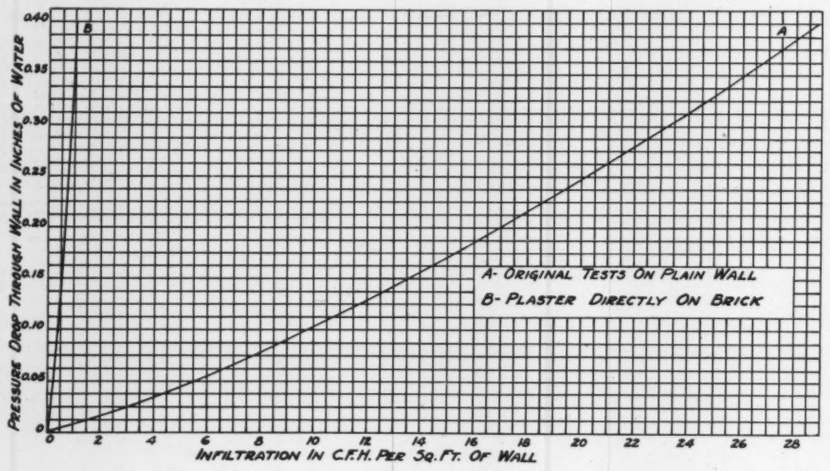


FIG. 9. INFILTRATION THROUGH WALL NO. 6—PLAIN AND PLASTERED

Three tests were made of the plastered wall equipped with the baseboard. In test run *A*, the baseboard was fitted as closely as could be against the sand finished plaster. In this case, the end and bottom joints between the baseboard and wall were completely sealed. The baseboard was routed out or relieved in the center so that the bearing on the wall was over a width of 1¼ in. at the top and bottom. In test run *A*, the fit to the wall was such that at no place could a 1/32-in. gage be slipped between the baseboard and the plaster.

It is estimated that the average clearance was almost 1/32 in., allowing for the roughness of the sand-finished plaster. In test run *B*, the baseboard was completely sealed on the top as well as on the bottom and ends. This corresponds then fairly closely to a test of the wall when completely plastered and sealed at the joint against the steel frame members. In test run *C*, the baseboard was removed completely and the furring space was open to the collecting chamber or room side of the testing machine through the inch or two of irregular space at the bottom of the plaster sheet. This run then corresponds to a test of the plain wall not plastered, since all parts of the brick and mortar surface have access to the furring space.

Fig. 11 shows the results of these three test runs in graphical form. The results are given in total infiltration through the wall in cubic feet per minute. The difference between the values on Curves *C* and *B* gives the reduction in leakage that is due to plastering completely, allowing no leakage around the baseboard. The difference between the values on curves *C* and *A* shows the reduction in leakage occurring when the plaster is not perfectly sealed at the bottom and a baseboard is applied. The reduction in leakage for plastering completely (*C*-*B*) is 2.27 cu. ft. per minute at 15 miles per hour for the entire wall. The reduction in leakage over that of the plain brick wall for the condition where the plaster is not perfectly sealed at the baseboard is $C - A$, equal to 1.06 cu. ft. per minute at 15 miles per hour for the entire wall. The effectiveness of the imperfectly sealed plaster and baseboard is then measured by the ratio of $\frac{C - A}{C - B}$, which is about 47 per cent. Then, the plaster and baseboard as built reduced the leakage roughly one-half as much as would the best plaster coat. This seems to bear out the importance of sealing the edges of the plaster sheet as at the baseboard, if the value of plaster in stopping air infiltration is to be realized in building construction.

To study the relative importance of infiltration through brick walls as compared to other factors involved in heat loss calculations, the following example has been worked out. The comparison was made on a ten-story office building omitting the basement. The total boiler load figured 12,200 sq. ft. of direct radiation for a temperature difference of 70 deg. and an emissivity of 240. The wall area, exclusive of window and door openings, was 37,860 sq. ft. and in this example was considered as all 13-in. brick wall with furring space, metal lath and ¾-in. plaster. The infiltration factors used on the windows and doors averaged about 110 B.t.u. per hour per foot of crack.

An effectiveness of plaster of 50 per cent was used in accordance with the baseboard tests on Wall 7 included in this paper. For Wall 4, this would be an infiltration value of 8.46 cu. ft. per square foot per hour, or a heat loss factor of 10.7 B.t.u. per square foot per hour for 70 deg. temperature difference at 15 miles per hour. For Wall 5, the corresponding values are 2.00 cu. ft. and 2.5 B.t.u. Assuming one-half of the wall area exposed to wind at one time, the calculated radiation to overcome infiltration through the brick on Wall 4 would be 926 sq. ft. or 7.6 per cent of the total calculated radiation, and for Wall 5 would be 220 sq. ft., or 1.8 per cent of the total calculated radiation.

The infiltration heat loss through the brick, based on the results of tests on Wall 4, was 25 per cent of the infiltration through windows and doors, and the corresponding figure for Wall 5 is 6 per cent.

The comparisons probably represent conditions in a building where care has not been taken in sealing the

plaster sheets at the baseboards and other wood trim. For a case where the sealing of the plaster is perfect, the amount of infiltration through the walls would be greatly reduced.

Based on the test results for Wall 4, the heat loss due to the infiltration through the walls would be 2.7 per cent of the infiltration loss through the windows and doors and the corresponding loss using the results of Wall 5 would be 0.5 per cent. Expressed in terms of the total heat losses for the building, the heat loss due to infiltration through the walls would be 0.8 per cent for the type of construction corresponding to Wall 4 and 0.14 per cent for Wall 5.

This indicates the importance of reducing infiltration for brick walls furred and plastered by the proper sealing of the plaster sheet at the baseboard and other wood trim.

Conclusions

Plain brick walls vary greatly with respect to air infiltration. Of the three factors—brick, mortar and workmanship—workmanship seems to be of most importance, the composition of the mortar as to cement and lime content next in importance, and the brick the least important. The infiltration for the 13-in. plain brick walls tested ranged from 2.71 to 10.35 cu. ft. per hour per square foot of walls.

Gypsum plaster, when properly applied, stops almost all infiltration. The results show that plastering stopped about 96 per cent of the leakage of the plain brick wall. Plastering directly on brick seemed to be slightly better than on metal lath with a furring space. The difference is small and the saving due to plastering directly on the brick would be negligible as compared to the saving in heat transmission effected by the furring space in the other construction.

The effectiveness of plaster in stopping infiltration in actual building construction is probably much less than that found in these tests. Cracking of the plaster, imperfect sealing of the plaster sheet at the wood trim, such as at the baseboard, would decrease the effectiveness greatly. A test made of a baseboard equipped wall, with a furring space, showed an effectiveness of only 50 per cent of that obtained with a perfectly sealed plaster wall.

Two ordinary coats of a linseed oil paint applied directly to the surface of a porous brick wall reduced the leakage by 9 per cent. A third coat applied with extreme care made the total reduction in infiltration 28 per cent.

One heavy coat of cold water paint applied directly to the surface of a hard brick wall reduced the leakage by 50 per cent.

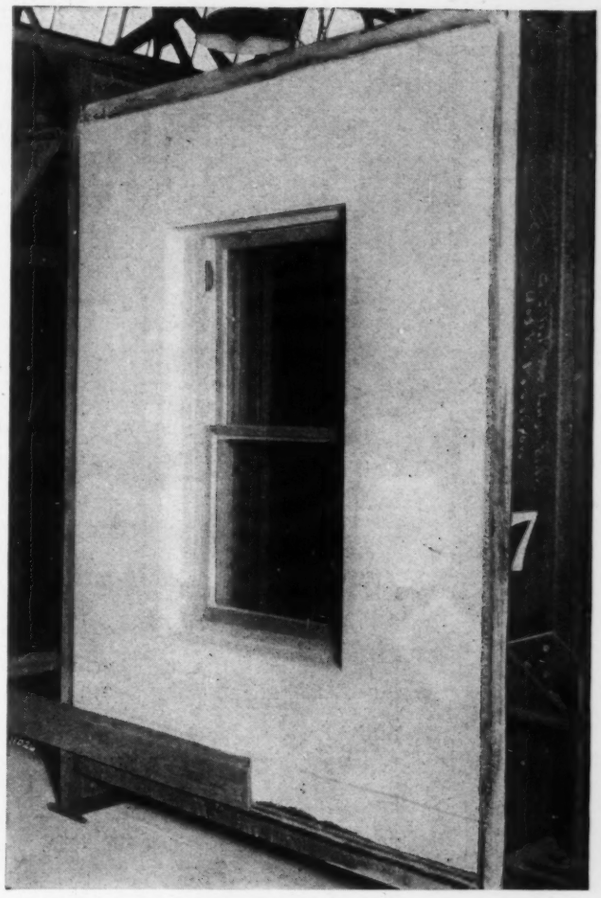


Fig. 10. Plaster and Baseboard Construction on Wall No. 7

A study made on a typical 10-story office building, considering all walls to be 13-in. brick with furring space, metal lath and plaster, indicates that the infiltration heat loss through the brick would range from 6 to 25 per cent of the infiltration heat loss through the windows and doors where the plaster is not properly sealed. For perfect sealing of the plaster, these percentages would be reduced to 0.5 to 2.7 respectively. This would indicate that the infiltration loss through brick walls with properly sealed plaster at the baseboard and other wood trim is negligible in making calculations for heat loss.

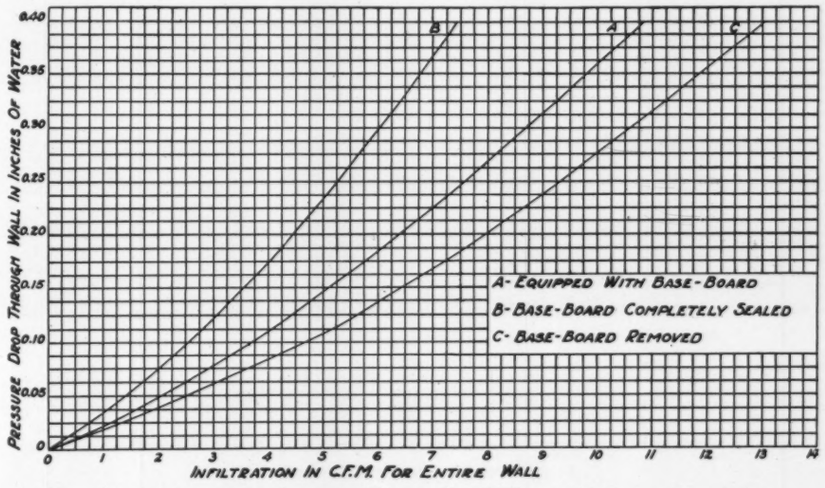


Fig. 11. Infiltration Through Wall No. 7 Under Various Conditions

PRESIDENT'S PAGE

THE members have started the campaign to increase the Society's membership wholeheartedly and it is splendid to see the enthusiasm of the individuals as well as the chapters in this very important work. Suitable rewards await the results of their efforts.

A new era in Society accomplishments is at hand, for the organization in its growing strength is reaching to wider fields, and a greater appreciation, particularly of research work, is noticeable. Because of the plans instituted by the Committee on Research through cooperative agreements, every dollar of members' dues allotted to research is made to do double and triple duty so that many more problems may be undertaken than would otherwise be possible.

Some conception of our research activities may be gained by Society members who have the opportunity to visit the laboratory November 4 when the annual open house is held.

Five definite problems are under investigation at the U. S. Bureau of Mines Experiment Station where our Laboratory is located and nine more are being handled in the engineering schools where cooperative agreements are in force. One of the worth while opportunities open to members is the chance to visit the Laboratory and this should not be neglected. The Society is admired for its research work and has gained in prestige since its laboratory results have been published.

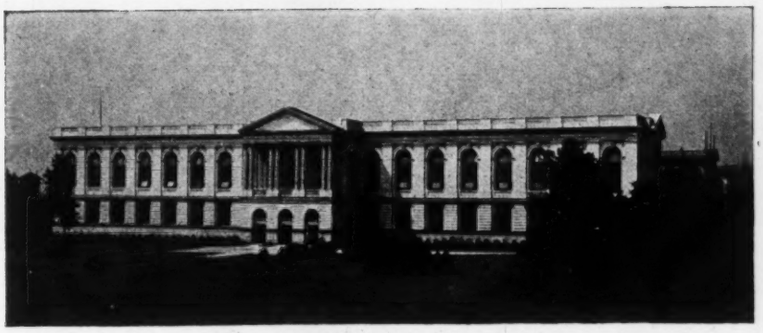
The next major function which should be attended by every member is the 36th Annual Meeting in Philadelphia, January 27-31 and the *International Heating and Ventilating Exposition* in the Commercial Museum during the same week.

In addition to a fine program of technical papers those who attend will meet men from all over the world that have a common interest. A fine display of modern equipment is in prospect and an added attraction will be the historical exhibit which will show the development of the industry.

It is a good time to consider the fine things that come to us through our membership in the Society. It is appropriate to ask ourselves what we have contributed to the profession of which we are a part.

Wouldn't it then be worth while to enter enthusiastically into every Society activity and be a working part of this organization?

Thomas Lewis



COMMERCIAL MUSEUM OF PHILADELPHIA

A Big Exposition in Philadelphia

Heat was first experienced by prehistoric man when he perceived fire as a phenomenon. Ventilation was not in vogue in those days, neither was air-conditioning. So the history of heat has come down to us through the ages as science has progressed and perfected its manifold uses. Ventilation is a product of the last decade and air conditioning is now passing through the growing stage but is rapidly rounding itself out into one of the most important professions in the country.

It is not so many years ago that the impression went forth of something subtle and secret in reference to the design of boilers and other heating apparatus which the manufacturer was not willing to give to the world—some curious or weird phenomena that people must not know about. Suddenly there was an awakening. The veil of mystery was lifted and engineering development entered upon a new era.

Illustrating this progressive spirit of modern manufacturing and merchandising efficiency, those men who have made history in the heating and ventilating field have pooled their interests and will give to the world for its closest scrutiny a new Declaration of Independence in Commercial Museum, Philadelphia, January 27 to 31, by placing on display at the International Heating and Ventilating Exposition the latest and greatest achievements in heating, ventilating and air-conditioning methods, machinery, material and equipment.

The Exposition will be held coincident with the annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Aerofin Corp.
Aerologist, The.
Aerovent Ventilating Co.
Air Control Corp. of New York.
Airmaster Corp.
Air Reduction Sales Co.
Alexander Bros., Inc.
Allen Air Turbine Ventilator Co.
American District Steam Co.
American Gas Association.
American Gas Products Corp.
American Metal Products Co.
American Oil Burner Assoc., Inc.
American Oil Burners Corp.
American Radiator Corp.
American Warming & Ventilating Co.
Ames Pump Co.
Armstrong Machine Works.
Automatic Burner Corp.
Barnes & Jones.
Beckwith Co.
Bishop & Babcock Sales Co.
B-Line Boiler Co.
Bristol Co., The.
Brown Instrument Co.
Bryan Steam Corp.
Bryant Heater & Mfg. Co.
Builders Iron Foundry.
Burnham Boiler Corp.
Carrier Engineering Corp.
Cashin Co., W. D.
Century Engineering Corp.
Celotex Co.
Champion Sales Co.
Cochrane Co.
Columbia Burner Co.
Combustion Fuel Oil Burner Co.
Combustion Specialties Corp.
Cooling & Air Conditioning Corp.
Cook Electric Co.
Copper Radiator Sales Corp.
Coppus Engineering Corp.
Crane Co.
Chicago Pump Co.
Cox, Abram Co.
Crystal Oil Burner Corp.
Culbert-Whitby Co.
Direct Control Valve Co.
Domestic Stoker Co.
Dunham Co., C. A.
Durston Gear Corp.
Economy Oil Burner Co.
Economy Pumping Machinery Co.
Engrg. Publications, Inc.
EverHot Heater Co.
Flagg & Co., Stanley G.
Frank Heater & Eng. Co., O. E.
Foxboro Co.
Fuel Oil & Temperature Journal.
General Electric Co.
General Gas Light Co.
Grinnell Co., Inc.
Hardinge Bros.
Hart & Crouse Co.
Hart & Hutchinson Co.
Hartzell Propeller Co.
Heating Journals, Inc.
Heating and Ventilating.
Heggie-Simplex Boiler Co.
Hill Company, E. Vernon.
Hoffman Specialty Co., Inc.
Hirschman Co., W. F.
Humphrey Radiant Fire Mantel, The.
Illinois Engineering Co.
Iona Ventilator Co., Inc.
International Burners Corp.
Janette Mfg. Co.
Johnson Company, S. T.
Johnson Service Co.
Kelly Brass Works.
Kewanee Boiler Corp.
Kieley & Mueller, Inc.
Korfund Co., Inc.
Langenberg Mfg. Co.
Lehigh Fan & Blower Co.
Leslie Co.
Linde Air Products Co.
Liquidometer Corp.
Marsh & Co., James P.
Massachusetts Blower Co.
McDonnell & Miller.
McIllvaine Burner Corp.
McQuay Radiator Corp.
Mears-Kane-Ofeldt, Inc.
Mercoid Corp.
Milwaukee Air Pump Co.
Minneapolis-Honeywell Reg. Co.
Mineral Felt Insulating Co.
Modine Mfg. Co.
Molby Boiler Co.
Moterstoker Corp.
Monarch Metal Weatherstrip Co.
McCord Radiator & Mfg. Co.
Multicell Radiator Co.
Meyer Furnace Co.
May Oil Burner Corp.
Mueller Furnace Co., L. J.
Nash Engineering Co.
National Regulator Co.
Nesbit, Inc., John J.
Newport Boiler Co.
National Radiator Co.
National Warm Air Heating Ass'n.
O. E. Specialty Mfg. Co.
Orr & Semtower, Inc.
Parks-Cramer Co.
Penn Electric Swtich Co.
Pennsylvania Furnace & Iron Co.
Pierce, Butler & Pierce Mfg. Corp.
Piqua Electric Mfg. Co.
Pittsburgh Water Heater Co.
Power Efficiency Corp.
Rawlplug Co., Inc.
Raymond Co., F. I.
Richmond Radiator Co.
Robertson Co., H. H.
Rome Brass Radiator Corp.
Roper Corp., George D.
Ric-Wil Co.
Richardson & Boynton Co.
Sarco Company, The.
Schade Valve Mfg. Co.
Sheffler-Gross Co., Inc.
Skinner Bros. Mfg. Co., Inc.
Silent Automatic Corp.
Skidmore Corp.
Smith Twin Tubular Boiler Co.
Spencer Heater Co.
Sprayo-Flake Co.
Star Expansion Bolt Co.
Stickle Steam Specialties Co.
Strandwitz & Scott, Inc.
Sweet & Doyle Foundry & Mach. Co.
Sterling Engineering Co.
Sheer Company, H. M.
Schutte & Koerting Co.
Staynew Filter Corp.
Taco Heaters, Inc.
Texo Sales Co., Inc.
Thatcher Co.
Time-O-Stat Controls Co.
Trerice Co., H. O.
Thermal Units Company.
Union Carbide Co.
United States Gypsum Co.
United States Radiator Co.
Vinco Co., Inc.
Warren Webster & Co.
Webster Tallmadge & Co., Inc.
Westinghouse Elec. & Mfg. Co.
Wilcolator Co.
Wilson Bros.
Winchester Repeating Arms Co.
Winslow Boiler & Engrg. Co.
Wood Conversion Co.
Wright Austin Co.
York Heating & Ventilating Co.
Young Radiator Co.
Young Pump Co.
Yale & Towne Mfg. Co.

INDEPENDENCE HALL AND LOWER BUSINESS DISTRICT OF PHILADELPHIA

The 36th Annual Meeting of A.S.H.V.E.

AN EXCEPTIONAL program has been prepared for the Society's 36th Annual Meeting to be held in the Benjamin Franklin Hotel, Philadelphia, Jan. 27-31, 1930, and it is anticipated that the largest attendance ever recorded at a Society function will be present when Pres. Thornton Lewis calls the first session to order.

After an eight year absence the Society will again enjoy the hospitality of Philadelphia and the local chapter members, under the capable leadership of John Cassell, Honorary Chairman, and R. C. Bolsinger, General Chairman, are cooperating to make this meeting an historic event.

The various committee chairmen have started on their appointed tasks and those responsible for the various activities are: Lee Nusbaum, Finance Committee Chairman; M. C. Gillett, Ladies Entertainment Committee Chairman; F. D. Mensing, Transportation and Publicity Committee Chairman and A. J. Nesbitt, Banquet Committee Chairman.

Hotel headquarters have been selected most appropriately, the Benjamin Franklin Hotel, named after the great American patriot and the Society's patron saint. The hotel is centrally located in the downtown business, shopping and theater district and here the social and business activities will center.

The first official function will be the meeting of the Council Monday, Jan. 27, and during the day a series of conferences of technical committees will be held. The first general meeting will be opened by President Lewis at 9:30 A. M., Tuesday, Jan. 28, and on succeeding days *morning sessions* will be held in the Benjamin Franklin Hotel and *afternoon sessions* beginning at 2:30 P. M. will be held at the Commercial Museum where the first International Heating and Ventilating Exposition is to be open during the entire week of Jan. 27-31.

With the combined attractions of a varied technical program, a large exhibition of modern equipment and a series of unusual social functions, there is an unique opportunity for Society members and their friends to spend a most enjoyable time in Philadelphia.

The Program Committee composed of H. H. Angus, Chairman, A. J. Nesbitt and Prof. F. B. Rowley has made an early start and announces a tentative list of subjects which are proposed for discussion at the 7 technical sessions:

Preventing Condensation on Interior Building Surfaces, by Paul D. Close.

Pressure Difference Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall.

Some Studies of Airation of Garages, by W. C. Randall and L. W. Leonhard.

Electric Power Generation by Process Steam, by L. A. Harding.

Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz.

Air Conditioning of the United States Capitol, by L. L. Lewis and A. E. Stacey.

Stokers for Domestic Heating, by R. V. Frost.

A Review of the Developments in Domestic Stoker Design, by Joseph Harrington.

Welding of Pipe for Heating Systems, by F. W. Johnson.

Automatic Controlling Devices for Heating Systems, by F. I. Raymond.

Electric Heating for Homes, Factories and Other Buildings, by W. S. Scott.

Measurement of the Flow of Air Through Registers and Grilles, by Davis.

Surface Transmission Coefficients, by F. B. Rowley.

Measurement of the Absorption of Heat from the Sun's Rays, A. S. H. V. E. Laboratory.

Wind Velocity Gradients Near a Wall, A. S. H. V. E. Laboratory.

This gives a program of wide scope and papers and reports will

be so arranged that plenty of opportunity will be given for discussion.

For the social part of the program some of the time-honored customs will be observed and several novel features have been added. As usual the officers and Council will entertain the authors of papers at luncheon on Monday. The Past Presidents will have their annual dinner and while they are dining their wives will have a special dinner party. A special program for the ladies in attendance is in charge of M. C. Gillett and it is assured that they will have no dull moments.

The major social event will be the annual banquet and dance at which the presentation of the past president's emblem will then be made. Several surprises at this dinner are promised.

In addition to the formal program Philadelphia possesses hundreds of places of historic interest as well as fine buildings and modern plants.

Time has been allotted not only for a visit to the exposition but also for side trips and tours of inspection. Bus service has been arranged for between the hotel and the Commercial Museum so that members can go to sessions or to see the exposition at convenient times.

With the preliminary arrangements all made, present indications are that the 36th Annual Meeting of the Society will be one that will make history.

National Fuels Meeting

An attendance of more than 1,000 at the Third National Fuels Meeting of the *American Society of Mechanical Engineers* at Philadelphia, October 7-10, 1929, was evidence of the great interest manifested in the program of popular and technical papers in which emphasis was placed upon domestic and industrial phases of fuel utilization, and upon smoke abatement.

The meeting was held in the Bellevue-Stratford Hotel, and was opened at 10:30 Monday morning with an address by the Hon. Harry A. Mackey, Mayor of Philadelphia. Conrad N. Lauer, vice-president-elect of the *American Society of Mechanical Engineers,* representing Pres. Elmer A. Sperry at the meeting, responded to the Mayor's greetings.

Dr. Robert H. Fernald, director of the Department of Engineering, University of Pennsylvania, was introduced by V. J. Azbe, chairman of the Fuels Division, for the presentation of his address on The Fuels Engineer, his Training and Work. Dr. Fernald lamented the weakness of many of our mechanical engineering curricula in the very important subject of fuel utilization and offered valuable suggestions for stimulating increased activity on the part of our educational institutions.

The following papers were presented at the meeting:

Economics of Pulverized Coal, Gas and Oil Fuels, Martin Frisch.

Sampling Pulverized Fuel, H. C. Porter.

Philfuels, R. W. Thomas.

Some Fundamentals of the Domestic Heating Problem, Samuel S. Wyer.

Stokers for Apartment Houses and Office Buildings, Joseph Harrington.

Economics of Reclamation of Anthracite Culm, Frederick H. Dechant.

Cleaning of Anthracite, T. M. Chance.

Coal Pretreatment, Clarence B. Wisner.

Symposium on Recent Developments in Mechanical Stoker Design. Presentation by H. D. Savage, President, *Stoker Manufacturers' Assn.,* Co-authors: R. A. Foresman, J. G. Worker and J. W. Armour.

Fuel Burning in Ceramic and Lime Industries, Victor J. Azbe.

Economics of Industrial Heating Practice, J. A. Doyle.

Use of Fuel in the Manufacture of Portland Cement, H. P. Reid.

Electricity for Industrial Heating Furnaces, W. S. Scott.

Gas for Industrial Heating Furnaces, H. O. Loebell.

Oil for Industrial Heating Furnaces, Leod D. Becker.

Carbonization Plant, "Lurgi" Process of Lehigh Briquetting Co., Max Toltz.

The Hayes Process Low-Temperature Carbonization Plant at Moundsville, W. Va., J. D. McQuade.

The Membership Work

October 15—January 15

The month of November gives promise of excellent results toward the increase of Society membership. Every member of the Society has received a letter from C. W. Farrar, Chairman of the Committee on Increase of Membership, telling of the plan to work for a substantial addition to the rolls between October 15 and January 15. More letters are coming.

The plan is to have each individual member lend his best efforts toward securing the applications of qualified men who are actively engaged in the heating and ventilating profession. Each chapter will cooperate through its Membership Committee in strengthening the local organizations.

As a reward for these efforts:

1. **The Society member whose name appears as first proposer on the greatest number of membership applications between October 15, 1929 and January 15, 1930, will receive a gold watch with Society emblem; for the second greatest number a copy of Vol. 2, Heating and Ventilation (revised) by Harding and Willard; for the third greatest number a pair of cuff links with Society emblem; for the fourth greatest number a watch fob with Society emblem and for the fifth greatest number a solid gold Society pin.**
2. **For the Chapter whose percentage of membership is increased by the greatest amount between October 15 and January 15 a banner will be awarded and will remain in possession of the chapter for one year.**
3. **Officers of the Society and members of the Council are not eligible for the prizes listed.**
4. **Awards are to be made at the banquet of the annual meeting in Philadelphia.**
5. **All applications must be properly filled in as required by the Society's Constitution and By-Laws and initiation fee check must be attached.**

Some Economic Considerations of Water Wall Installations, Ollison Craig.

Economics and Design of Water-Cooled Furnaces, J. S. Bennett and P. N. Oberholtzer.

Effect of Fouling in Boiler Efficiency, J. W. Pierson.

Symposium on Domestic Heating. Papers on: Economics of House Heating and Insulation, H. B. Johns. Gas, M. J. Roberts. Oil, S. D'Arcy Rickard. Electricity, A. R. Stevenson, Jr. Anthracite, A. F. Duemler. Bituminous, F. R. Wadleigh. Coke, P. Nicholls.

Smoke Abatement, Harvey N. Davis.

Smoke Abatement at Salt Lake City, J. Billeter.

Smoke Abatement at Nashville, Tennessee, George C. Fisher.

Smoke Abatement at Knoxville, Tennessee, F. L. Wilkinson, Jr.

Atmospheric Pollution and Sunlight, Philip Drinker.

Symposium on Methods of Recording Smoke Density. Papers by: A. S. Langsdorf, H. V. Breisky and Victor J. Azbe.

The domestic session held Wednesday evening was perhaps of greatest interest to the meeting. Designed primarily to acquaint the general public with some of the principal fuel problems and their possible solution, the meeting proved of tremendous interest to the engineers in attendance, as well as the non-technical men. Several papers were presented including a general discussion of economics of house heating and building insulation.

The smoke abatement session also proved quite valuable and records were submitted of the activities in cities in which smoke campaigns are being waged.

Instruments for the measurement of smoke and for photographing offending stacks were exhibited and attracted considerable attention.

A conference on coal research on Tuesday afternoon proved exceptionally valuable as a means of stimulating interest in this work. It appears that one of the first projects that should be undertaken is that of sampling pulverized coal.

A plant inspection program included some of the best known industrial establishments in the Philadelphia district. Among them were the Navy Yard, the Philip Carey Co. plant, the Valley Forge Cement Corp. plant, the heating plant of the Pennsylvania Railroad, the oil refining plant of the Atlantic Refining Co., the American Engineering Co., the Dill and Collins Co., a boat trip touching the Delaware River industries, Baldwin Locomotive Works, Aberfoyle Mfg. Co., and the Westinghouse Electric and Mfg. Co.

Five Chapters Visited by President Lewis

During the month of October President Thornton Lewis, Secretary A. V. Hutchinson and C. W. Farrar, Chairman of the Committee on Increase of Membership, visited the members of the Massachusetts Chapter in Boston October 8; Illinois Chapter in Chicago, October 14; St. Louis Chapter October 15; Kansas City Chapter, October 16, and Minnesota Chapter in Minneapolis, October 17.

President Lewis selected for his topic: The Future of the A. S. H. V. E. and Mr. Farrar outlined the details of a membership campaign to be conducted between October 15 and Jan. 15.

In each city the visitors were splendidly entertained by the council members and chapter officers. At St. Louis a visit was made to the National Dairy Exposition and in Kansas City the city's airports were inspected as well as the recently rebuilt plant of E. K. Campbell.

A rousing greeting was given by Minnesota in the form of a monster "pep-fest" just outside the meeting hall at the University of Minnesota.

On Monday, October 21, the Secretary visited the Wisconsin Chapter at Milwaukee and addressed the members on Society activities.

President Lewis and Mr. Farrar will appear at the November meeting in Milwaukee and will address the Pittsburgh chapter members November 4.

Local Chapter Reports

New York Chapter

The opening meeting of the 1928 season was held on October 15, which was given over to a general discussion on the recent progress in heating and ventilating.

The November meeting, which was held on the 12th of the month, proved to be most interesting. It consisted of an inspection trip through the Laboratory of the Institute of Technical Research of the American Radiator Co., Yonkers, N. Y. Following the dinner, which was served at the laboratory, the regular order of business was carried on. Dr. Brabbee then gave a very interesting talk on the subject of radiators, humidity and boilers.

Prof. L. E. Seeley of the Yale Scientific School, New Haven, selected the Rating of Heating Boilers as his topic for discussion at the December 17th meeting.

The January meeting was given over to the lecture on the West Street Telephone Bldg., which had been scheduled for the October meeting.

Experiences in burning coke was the subject discussed at the meeting held on February 18th.

At the March meeting A. W. Pihlman gave a talk on the recent developments in house heating by gas which brought about an interesting discussion.

An inspection trip through the plant of the Carrier Engineering Corp. of Newark, N. J., was arranged for the April 15th meeting which was enjoyed by those members and guests who attended the meeting. A dinner was served at the plant, which was followed by an address given by J. I. Lyle on the subject of heating and ventilating the Houses of Congress at Washington, D. C.

The last meeting of the season was turned into a social with special talent arranged by the Entertainment Committee.

The officers and board of directors now serving the New York Chapter are as follows:

President, E. J. Ritchie

Vice-President, A. J. Offner.

Treasurer, F. E. W. Beebe.

Secretary, E. B. Johnson.

Board of Governors, H. B. Hedges, C. W. Brabbee and Russell Donnelly.

Philadelphia Chapter

October 10, 1929: At the October meeting of the Philadelphia Chapter a moving picture, entitled The Wonders of Anthracite, was presented to the members present by Mr. Hansberger of the Lehigh Coal & Navigation Co. At the conclusion of the picture a rising vote of thanks was extended to Mr. Hansberger and his company, on motion of A. J. Nesbitt, seconded by J. D. Cassell.

The minutes of the preceding meeting and the treasurer's report were read and approved, and the names of six members, four Junior members and four Limited Chapter members were announced at this time.

M. F. Blankin, chairman of the Meetings Committee, outlined the tentative schedule of meetings for the next few months. After an interesting discussion by Messrs. Bolsinger, Mensing

and Nesbitt of the proposed Ladies' Night, it was voted that the December meeting be designated as such, and all members were urged to obtain a good attendance for the occasion.

J. D. Cassell read a communication from H. P. Gant, chairman of the International Heating and Ventilating Exposition, to be held in Philadelphia during the week of January 27, in regard to the proposed collection of old and obsolete types of heating and ventilating equipment and tools. The committee desires to establish a museum section for the exposition, contrasting the old types of equipment with the modern types. Mr. Cassell approved the suggestion of the committee and urged all members to be on the lookout for suitable equipment. The necessary transportation will be furnished for moving such apparatus to the Commercial Museum.

There being no further business, the meeting adjourned.

Wisconsin Chapter

September 30, 1929: The first meeting of the Wisconsin Chapter for the current season was held on September 30, with an attendance of 19 members and guests.

Following the regular order of business the new officers and board of directors elected to serve for the coming season are as follows:

President, Fred G. Weimer.
Vice President, John G. Shodron.
Secretary, V. A. Berghoefer.
Treasurer, Martin Erickson.
Board of Governors, Paul C. Downey, Prof. G. L. Larson and W. F. Noll.

After the election of officers, plans for the coming year were discussed, which was followed by adjournment.

New Offices for M. G. Harbula

M. G. Harbula, member of the American Society of Heating and Ventilating Engineers, who has been specializing in Air Conditioning for over fifteen years, is now engaged in specializing in air conditioning consulting engineering. Mr Harbula has recently announced the removal of offices to Suite 846, General Motors Bldg., 1775 Broadway, New York City.

Revised Edition of Harding and Willard Released

The new edition of "Mechanical Equipment of Buildings, Volume I" by L. A. Harding and A. C. Willard, has recently been published by John Wiley and Sons, New York.

This is one of the most widely-used reference books available on heating and ventilation, and has been extensively used as a text book in many universities since publication of the first issue in 1916.

The developments in the field of heating and ventilation since 1916 have been remarkable. Scientific research, invention and manufacturing have produced new and improved equipment, all of which have materially altered the practice of thirteen years ago. It was therefore apparent to the authors that a complete revision of this volume was necessary in order to bring it into agreement with the present state of the art. In making this revision, they not only added new material to the present chapters, but they also carefully replaced obsolete material in each chapter with the best and latest data available.

Practically every chapter in this volume contains new material, and certain chapters have been entirely rewritten. The book contains numerous charts and illustrations.

Acknowledgment is made by the authors of the many valuable suggestions of the late Prof. G. A. Goodenough, as well as permission to make use of his latest tables of the properties of steam and ammonia, and also of air and vapor mixtures. Profs. A. P. Kratz, W. H. Severns and V. S. Day also contributed technical material for the second edition of this volume.

The authors of this book are well known to members of the American Society of Heating and Ventilating Engineers, as well as the engineering world at large. Professor Willard is a past president of the Society, and is now a member of the Council. He is head of the Department of Mechanical Engineering, University of Illinois, and was Consultant on Ventilation for the Holland Tunnel. Mr. Harding is now Vice-President of the Society, and was formerly Professor of Mechanical Engineering at Pennsylvania State College. He is a contributor to Kent's Mechanical Engineers' Handbook, Kidder's Architects' and Builders' Handbook, and is president of the L. A. Harding Construction Co., Buffalo, N. Y.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 26 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by December 1, 1929, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Barry, Curtis P., Narowetz Heating & Ventilating Co., Chicago, Ill.	L. Narowetz, Jr. E. P. Heckle	W. J. Uhlhorn
Belt, Newton O., Asst. Chief Engr., Armstrong Cork Co., Lancaster, Pa.	L. C. Davidson A. P. Shanklin	F. M. Holbrook F. P. Anderson

BRAKENRIDGE, CHAS. E., Anthracite Coal Service, Chicago, Ill.	Homer R. Linn H. G. Kreissl	C. M. Terry Lee G. Bird
CARRIER, EARL GARDNER, Carrier Engrg. Corp., Newark, N. J.	G. L. Larson L. L. Lewis	V. S. Day A. E. Stacey, Jr.
CHERNE, REALTO E., Carrier Engrg. Corp., Newark, N. J.	F. B. Rowley J. V. Martenis	J. I. Lyle L. L. Lewis
CRESSY, RALPH E., Sales Engr., Hoffman Specialty Co., Detroit, Mich.	L. A. Harding C. V. Haynes	M. C. Gillett Raymond E. Jones
EILBECK, ARTHUR BLAKE, Carrier Engrg. Corp., Newark, N. J.	W. H. Carrier J. I. Lyle	W. G. Hillen H. Worsham
ERNST, CHAS. EDSON, Carrier Engrg. Corp., Newark, N. J.	W. H. Carrier J. I. Lyle	W. G. Hillen H. Worsham
GERSTUNG, G. WALTER, Multicell Radiator Corp., Buffalo, N. Y.	H. F. Hutzel R. G. Thornton	L. A. Harding W. G. Fraser
HARRIS, CHAS. RUSSELL, Carrier Engrg. Corp., Newark, N. J.	W. S. Moffett W. H. Carrier	A. E. Stacey, Jr. J. I. Lyle
KAGEY, ISAAC BENJAMIN, JR., Hotel Winfield Scott, Elizabeth, N. J.	J. I. Lyle W. H. Carrier	E. T. Murphy L. L. Lewis
LYON, P. S., Chief Engr., Hall Elec. Heating Co., Inc., Philadelphia, Pa.	L. C. Davidson R. F. Hunger	A. P. Shanklin Geo. W. Barr
MARTINEZ, JUAN J., Asst. Supt., Mech. Dept., The Mexican Light & Power Co., Mexico City, Mex.	H. Diederichs D. S. Kimball	A. C. Davis M. C. Ernsberger
MATHER, HARRY H., Phila. Elec. Co., Philadelphia, Pa.	L. C. Davidson W. A. Bornemann	P. L. Davidson R. F. Hunger
MCGREGOR, ROBT. FLEMING, Carrier Engrg. Corp., Newark, N. J.	J. E. Cleland E. C. Wiley	J. I. Lyle A. E. Stacey, Jr.
MILLARD, JUNIUS W., Dist. Mgr., York Htg. & Vtg. Corp., Philadelphia, Pa.	J. V. Cavileer Thornton Lewis	Donald French J. A. Carey
NELSON, CHESTER L., Carrier Engrg. Corp., Newark, N. J.	F. B. Rowley W. H. Carrier	W. G. Hillen J. I. Lyle
MUSSON, JOSEPH R., Charge of Sales (Radiator Div.), Heintz Mfg. Co., Philadelphia, Pa.	Alan A. Miller Frederick D. Mensing	H. M. Patrick E. Nute Sanbern
SHEPARD, JOHN DEBERARD, Carrier Engrg. Corp., Newark, N. J.	W. H. Carrier J. I. Lyle	H. Worsham E. T. Murphy
PAYSON, CHAS. H., Owen, Saylor & Payson, Kansas City. Mo.	John H. Kitchen Herman Henrici	Carl Clegg Ben F. Natkin
SMITH, ELMER GILLAM, ASST. Prof. of Physics, College Station, Tex.	F. E. Giesecke	
THORNBURG, HAROLD ANDREW, Carrier Engrg. Corp., Newark, N. J.	F. Paul Anderson L. S. O'Bannon	W. H. Carrier J. I. Lyle
WACHS, LOUIS JOSEPH, Carrier Engrg. Corp., Newark, N. J.	F. Paul Anderson L. S. O'Fannon	J. I. Lyle W. G. Hillen
WILSON, ANDREW W., Sales Engr., McKellar & Blackhall, Toronto, Ont., Canada.	W. R. Blackhall J. S. Paterson	J. J. McKenzie W. Sheppard
WRIGHT, M. BIRNEY, Instr., Case School of Applied Science, Cleveland, Ohio.	M. S. Titus D. M. Quay	C. F. Eveleth W. C. Kammerer
YEOMANS, PAUL H., Armstrong Cork Co., Lancaster, Pa.	L. J. Klaus D. F. Patterson	R. B. Beahm, 2nd K. F. Rugart

Candidates Elected November, 1929

In the past issues of the Journal of the Society there were posted among the list of Candidates for Membership, 10 candidates for election in the different grades of Membership, whose names have been balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

BROOKS, SAMUEL J., Ch. Engr., Spencer Heater Co., 68 Adelaide St., East Toronto, Ont., Canada.

GRAHAM, WILLIAM D., York Heating & Ventilating Corp., 205 W. Wacker Drive, Chicago, Ill. (*Advancement.*)

HOOPER, VERNON FOWLER, Heating Engr., Lord & Burnham Co., Irvington, N. Y.

PARK, CLIFTON D., Dist. Sales Rep., Carrier Engrg. Corp., 1011 Statler Bldg., Boston, Mass.

VALE, HENRY A. L., Managing Director, Vale & Co., Ltd., 141-143 Armagh St., Christchurch, New Zealand.

ASSOCIATES

CRANDALL, R. KAY, Buick Motor Co., Flint, Mich.

SIMMS, ROBERT C., Dist. Supt. of Constr., Carrier Engrg. Corp. 1724 Land Title Bldg., Philadelphia, Pa.

WAITE, HARRY, Secy. & Mgr., Gray Plumbing Co., 1214 Ogden Ave., Superior, Wis.

JUNIORS

PETERSEN, ADOLPH J., U. S. S. Wilmette, Foot of Randolph St., Chicago, Ill.

WILLIAMS, DAVID BUTLER, Draftsman, Carrier Engrg. Corp., 1724 Land Title Bldg., Philadelphia, Pa.

Heating · Piping
and Air Conditioning

EDITORIALS

Power Piping Systems

Section two of the recommended code for pressure piping, on power piping systems, has recently been distributed to the industry for critical review. It represents another step in the gradual process of evolving a piping code of material specifications and standard dimensions for pipes, fittings, and valves, together with general requirements for hangers, coverings, inspections, tests, etc.

This particular section covers the design, manufacture, test and installation of power piping systems in steam generating stations, central heating plants, industrial process work and all distributing systems where the pressure is in excess of 15 lb. Other sections will cover gas and air piping, hydraulic piping, oil piping, refrigerating piping, piping materials and fabrication details. If and when all these sections are prepared and approved, they will become a national code for pressure piping.

Demand for such a code has developed gradually, fostered to some extent by the activities of the sectional committee on the standardization of pipe flanges and fittings and the work of the industrial commission of the state of Ohio in 1924. The Ohio piping code particularly stimulated action on a national code if only for the reason that if there were to be codes it would be better to have one of national scope. Therefore, the *American Society of Mechanical Engineers* was asked to initiate a movement for such a code under the procedure of the *American Standards Association.*

The *A. S. M. E.* accepted the invitation and held a preliminary conference in Milwaukee on May 19, 1925. Opinions at this conference were all favorable to the project, so in September of the same year the *A. S. A.* was requested to authorize the organization of a sectional committee to prepare it. In succeeding months this permission was granted, sixty-one national organizations were invited to appoint representatives to the sectional committee and sub-committees were formed to prepare the several sections of the code.

Certainly such a background of thought and action indicates the importance of a code for pressure piping in the opinion of many leaders in the industry. Therefore, it should receive the study of everyone in the industry in the effect it will have on piping practice. As it now stands, three of the sections, of which power piping is one, have been prepared and distributed for review. They have all been distributed in tentative form, for their committees want suggestions from the field before making final drafts to submit for the industry's approval.

Thus, it is now and as these sections appear that those in the industry should take active interest in the movement, so that whatever will result will represent the best of combined thought and experience.

Idle Equipment

The census of manufactures for 1929 will call for information on the amount of idle power equipment in manufacturing plants throughout the country. Definite data on this subject have never been secured, although unofficial estimates have been made that as much as 50 per cent of the installed power in the United States is practically idle through obsolescence, overcapacity or other causes.

This is information noted in a bulletin of the Department of Commerce.

Data of this sort should be interesting. The census will undoubtedly show a lot of equipment idle because plants have sensibly called it obsolete in favor of new, improved equipment. It will undoubtedly show equipment that is idle because it has been installed to take care of future expansion. At the same time, though, it will show plenty of equipment idle because the proper engineering calculations were not made at the time of original design and installation—and it is that kind of idleness that is not good business.

Take piping. That is the part of power equipment in which our readers are interested. Proper design of piping will reflect a decided economy in fuel or pumping costs over improper design. Yet industrial plants are piping steam, air, refrigerants, etc., etc. in lines that are over-sized and that are, therefore, partly idle due to overcapacity.

Such a census as is referred to will probably not show this. It will probably not consider the fact that pipe lines in use are partly idle because they are oversized. That is something for the piping engineer and contractor to consider. We hope that the proposed census will at least develop a consciousness of waste in plant equipment and operation that will cause such consideration.

Migration of Industry

Migration of industry will be a subject of inquiry for the first time in the coming census of manufactures. Movements of manufacturing which have been under way during the last decade have resulted in decreases in the requirements for skilled labor in one section of the country accompanied by increases in requirements for the same type of labor in other parts of the country. Some of these movements have entirely changed the industrial occupation of a large portion of the inhabitants of these sections. There has been a tendency toward concentration of some industries, while other industries have tended to spread throughout the country. And all these movements have their economic consequences—chiefly advantageous.

One of the contributing factors in the migration of industry is the freedom from climatic demands that air conditioning has made possible. Before the development and perfection of air conditioning it was necessary for many industries to settle in those portions of the country in which climatic conditions were favorable to their manufacturing processes. This might mean that such industries were forced to operate far from the sources of supply for their raw materials or, in general, to sacrifice possible economies to the need for certain humidities and certain temperatures which nature provided only in one place.

Not so now, with air conditioning prepared to produce whatever atmospheric conditions are required. If there be economic advantages for industry to move north, east, south or west or to move to urban or rural communities, it can do it. Weather is no longer a consideration.

Research

When President Hoover, along with the rest of the world, paid tribute to Thomas A. Edison for his discovery of the incandescent lamp, he paid even greater tribute to him for his method of discovery—research. He paid tribute to research because the same kind of research that gave the world the electric light has given the world countless other devices that have benefited mankind. And the recognition of Edison's contributions can well be interpreted as recognition of all research workers' efforts.

In the engineering world we are all working with principles and aids that have come to us from the research laboratory. It was fitting, therefore, that President Hoover, appreciative of the importance of research, took the occasion at the Edison ceremonies to bring to all the realization that there are other Edisons and that there should be more.

Quoting another president, Thornton Lewis of the American Society of Heating and Ventilating Engineers, an organization that has always fostered research, we ask each reader how much he would know, how much he would have to work with, if his knowledge and his materials were only those that he developed himself.

Yes, there is no doubt of our dependence upon research. And, considering what it has done for us, there is little doubt of the practicability of sponsoring more and more of it.

The World Studies Air Conditioning

The World Engineering Congress in Tokio, Japan, has rendered the air conditioning industry a signal service. Its importance has served to bring together some of the industry's foremost engineers—Carrier, Willard, Fleisher, Lewis, Still, Armspach, Gant, Yaglou, Ellis, Houghten and Harding—to prepare a paper on "The Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort." Needless to say, with such able thought, knowledge and experience contributing to it, this paper is a document of outstanding value to the industry.

For presentation to an international audience, it is international in character. It is a summation of accepted laws of air conditioning and of all methods of air conditioning. All those who have followed the development of air conditioning, who have been anxious to study air conditioning and who have had to find here and there bits of knowledge to make up their whole understanding of the process and its applications, have in this paper the "text book" they have long needed.

It brings air conditioning up to today. It enables air conditioning to proceed from today's status in its further development. Just released, it appears in this journal and, simultaneously, in the technical press throughout the world. It should become a reference work in the library of every engineer and contractor.

It is a matter of pride to think that the development of air conditioning is so largely an American development. In Japan at this world congress the new and the best in engineering will be presented and discussed, and in some phases of engineering the representatives of other countries will bring messages of advancement that is not yet paralleled here in America. But not so in air conditioning.

Air conditioning is international. It has installations in practically every country. But these installations as a whole represent American invention and American manufacture. It appears that we lead in this branch of engineering.

In this country air conditioning is serving over two hundred different industries. It is contributing to human comfort and health in the home, the theater, the department store, the club, etc., etc. Since its inception some twenty-five years ago, its sales volume has doubled on the average of every five years.

INTERVIEWS OF INTEREST

Carbon Monoxide Poisoning in Young Children

A very effective piece of practical health endeavor was initiated when Dr. Albert M. Stevens of New York City set about studying the gradual, cumulative effects of exposure of very young children to small but repeated amounts of carbon monoxide gas in the air they breathe. Dr. Stevens is associated with the department of diseases of children, Columbia University. He has abundant opportunity for investigative work in his special field. His connection with the children's medical division of Bellevue hospital brought to his notice the anemic type of city child which he has come to designate as a chronic condition due to the cumulative poisoning effect of repeated exposure to relatively small amounts of carbon monoxide gas. His experience is interesting:

"The children's division of Bellevue hospital has cared for seven cases of carbon monoxide poisoning in children during the last four years. One of them was fatal," states Dr. Stevens. "Furthermore, a careful study among child groups served by the hospital is rather convincing that still larger numbers of cases of chronic poisoning among children can be sifted out from among school groups. In fact, they are very easy to discover, once one's attention has been called to the typical signs of this disease.

"They are found among the children with uncertain health, who catch cold easily. They slump in school work. They are languid, anemic, and otherwise dull. Their appearance is so characteristic that on the basis of looks alone an initiated person can pick them out. They occur mostly in neighborhoods where small bedrooms depend upon gas for heat, and where old tubes and makeshift apparatus are in common use.

"Our one fatal case was an infant, whose crib had stood at night beneath an outworn gas fixture. The air it breathed was vitiated by very small but constant leakage of gas. The mother had suffered from morning headaches and other members of the family showed some effect, but the infant was the only member of the family continuously exposed to gas.

"The infant was very sick when we first saw it, but the high percentage of CO in the blood gave its body a pinkish flush that resulted in a quite deceptive appearance of health. It was this pinkish flush that led us to test for CO poisoning and resulted in the discovery of a blood concentration of carbon monoxide of over 18 per cent. The patient died before measures could be carried out for its control.

"The experience was timely. It disproves the long accepted statement that carbon monoxide is not a cumulative poison. It emphasizes that infants and young children are peculiarly susceptible to gradual poisoning from carbon monoxide gas. It points out the menace of chronic CO poisoning within the home. It suggests that extra precautions be taken to exclude the gas and to insure adequate ventilation of homes in winter, when seasonal increase of carbon monoxide menace is always to be expected."

Dr. Stevens describes at some length the symptoms of carbon monoxide poisoning:

In infants, the pulse is feeble, the respiration slow, the temperature normal or somewhat low.

In older children there is general pallor, there may be slight jaundice. There are headaches, breathlessness, and dilatation of the veins of the neck. The red blood cells and hemoglobin are low. Listlessness, progressive pallor, loss of appetite, fatigue, and disturbances of sight and hearing may occur. The milder cases of carbon monoxide poisoning are not easy to detect. Serious cases of poisoning will show blood concentration of carbon monoxide higher than 10 per cent.

CO_2 is a stable compound, harmless in small amounts and of much use in our physiological scheme of things. CO is an unsatisfied, greedy combination, on the lookout for another molecule of oxygen. The CO molecule attaches itself to the oxygen bearing cells in the blood stream and will not be dislodged. It satisfies its voracious demand for additional oxygen at the expense of the tissues, and tissue asphyxiation results.

As to the amounts of the gas inhaled before any perceptible symptoms are produced, Dr. Stevens states that 3 parts of CO gas per 10,000 parts of air produce no appreciable effect. That is to say, the body is able to overcome any effects of poison in this amount. In proportions of 6 parts per 10,000 physiological disturbance becomes perceptible. That is, if the gas is inhaled in amounts above 3 parts per 10,000 of air, there results some degree of poisoning. A ratio of 9 parts per 10,000 is sufficient to cause headache and nausea in workers, and 15 parts per 10,000 constitute an industrial menace.

Certain susceptible individuals may show earlier and more severe symptoms than others. One does not become inured to even very small amounts of the poison, and recent experiments go to show that the breathing of air containing as little as one part of carbon monoxide per 10,000 parts of air over a period of four hours by a person sitting at rest and breathing quietly produces the same effect as exposure to 4 parts of the gas for one hour.

The symptoms of chronic carbon monoxide poisoning in adults are, first of all, the occurrence of fatigue and muscular weakness; then headache, dullness, mental depression, poor memory, restricted fields of vision, ashy pallor, palpitation, irregularly of the heart, and, finally, nausea, vertigo, slowing of the pulse, dyspnea, dilatation of the veins of the neck, an increase in the globular elements in the blood, increasing torpor, death.

Exhaust Gases

Six volunteer students at the University of Pittsburgh have just undergone tests which worked out by actual experiment the safe limits of daily exposure to carbon monoxide gas for periods of four to seven hours to which New York policemen are subjected when on duty in Holland tubes.

A sun parlor especially equipped was used for the observations and tables were constructed of the physiological reaction of the men to varying periods of exposure to controlled amounts of exhaust gases, and after sixty days it is scientifically demonstrated that air in the tubes should have less than 2 parts per 10,000 parts of air, with working periods of from two to seven hours.

The Liberty tubes at Pittsburgh and the Holland tubes beneath the Hudson, New York, are kept below the danger point by special ventilation apparatus developed at Pittsburgh. The dizziness sometimes complained of by policemen in the tunnels is due to watching the automobiles pass, states the report.

Joint Committee on Atmospheric Pollution

Dr. Alexander Silverman, head of the department of chemistry, University of Pittsburgh, is well-fitted for his position as chairman of the committee on atmospheric pollution by automobile exhaust gases. His graduation thesis was on carbon dioxide and in the course of its development he made many relevant observations on carbon monoxide. Also he developed in the course of this work the energetic prosecution of field inquiry that is responsible for his many-sidedness now and the perennial up-to-dateness of the problems that concern him.

The joint committee is sponsored by the *American Chemical Society* and under the direction of Dr. Silverman the personnel of the committee is highly representative of allied interests, and committee members are technically competent. It is in fact a public relations council on the increasingly important hazard of carbon monoxide gas in highly urbanized communities.

"Improvement in ventilation as a whole cannot take place spectacularly," declares Dr. Silverman; "the problems are too numerous, the remedies too complex. Protective legislation, however useful, must be regarded as a temporary form of control during the interval when protective apparatus undergoes study and change, and installations become intelligently commercialized. Demonstration and educational effort do the rest. The public demands tomorrow a precision in these matters which we find it highly difficult to enforce today.

"In the matter of carbon monoxide pollution of the air we breathe, practically every city dweller suffers from some degree of chronic anemia from this cause. It is rare to find a city dweller whose blood count is up to 85 per cent. Drugs are available which bring this count up to 95 per cent and better, but the population as a whole requires other protection.

"If no other remedy can be found for the carbon monoxide evil, our large cities may require that all motor vehicles have an auxiliary electrical driving system for use in cities, limiting the use of gasoline to open country. The pernicious practice of keeping motors running while parking or delivering goods represents needless pollution. It constitutes needless waste. Thousands of dollars are wasted daily in every large city from this cause, which is preventable, alone.

"On the scientific side, researches now under way promise the perfection of a catalytic process by which automobile exhaust will be converted into the harmless CO_2, a simple thing that may revolutionize the ventilation arrangements of enclosed spaces where this gas is a menace. The trouble would then be controllable at its source.

"The menace of carbon monoxide in the home requires much thought. Forced air exchange is the price of safety in many domestic situations. Hardly any gas stove is adjusted to the burden it carries; and laboratories in many research institutions conduct operations with entire disregard of the extreme toxicity of the combustion products of such substances as nitrocellulose, for instance.

Ventilation of Laboratories

"The chemical laboratory is well served by a unit ventilation system. My own idea of an effective ventilation scheme for the chemical laboratory is one that will permit economical continuous use of any part of the building to meet the requirements of any type of experiment that may be set up. Usually this means some form of unit ventilation.

"Unit exhaust is essential, a hood with separate individual exhaust fan wherever noxious fumes or dusts arise, with no intercommunicating exhaust ducts anywhere.

"The air supply in my scheme would be by means of unit ventilations placed at ends of corridors. The air would be constantly supplied at sufficient pressures to assure positive pressure, a directional system, constant air currents, with windows as exhaust vents, not the source of supply.

"Ozone has important applications in ventilation problems of the chemical laboratory. Indeed, the chief development of ozone in air supply has been within the chemical industry.

"Committees at best are clumsy devices on which to hang our hopes for reform in ventilation practice. They serve to gather information and to set new problems of research, but the one quick route to educational broadcast is commercial interest, plus preparedness and individual enthusiasm.

"Attitudes have changed of late. My earliest efforts at field inquiry were baffled on every hand by individual and industrial opposition. Industries were afraid to be 'investigated.' Now it is different. Associations are formed to elicit all the facts, and earlier non-co-operative groups finance foundations to carry out essential projects in research. Industry realizes that it must put its own house in order with respect to air pollution, and proposes to keep the initiative where it in fact belongs.

Progress Made

"Specific progress has been made. The work of W. P. Yant, himself a member of the joint committee, has developed recording instruments so delicate that it keeps continuous charts of carbon monoxide in tunnels and other similar danger points so that it registers the gas

automatically and rings a bell when concentration reaches a point as high as 1 part per 10,000 parts of air.

"The technical competence of joint committee members as well as the industrial representation of its membership assure its efficient future work. Reports are current that a new cheap gas for household use has the merit of being non-asphyxiating also. Progress is certain, but reform is slow. Ten years at least of earnest effort will be required to make improved standards universal."

Other members of the committee include Dr. H. C. Dickinson, American Automobile Association; Dr. Paul N. Leech, American Medical Association; R. P. Anderson, American Petroleum Institute; Dr. George A. Soper, American Public Health Association; T. D. Pratt, Motor Truck Association of America; Walter S. Laird, National Association of Taxicab Owners; Dr. Leonard Greenburg, National Safety Council; T. A. Boyd, Society of Automotive Engineers; W. P. Yant, United States Bureau of Mines.

Control of Humidity and Temperature

(*Continued from Page 543*)

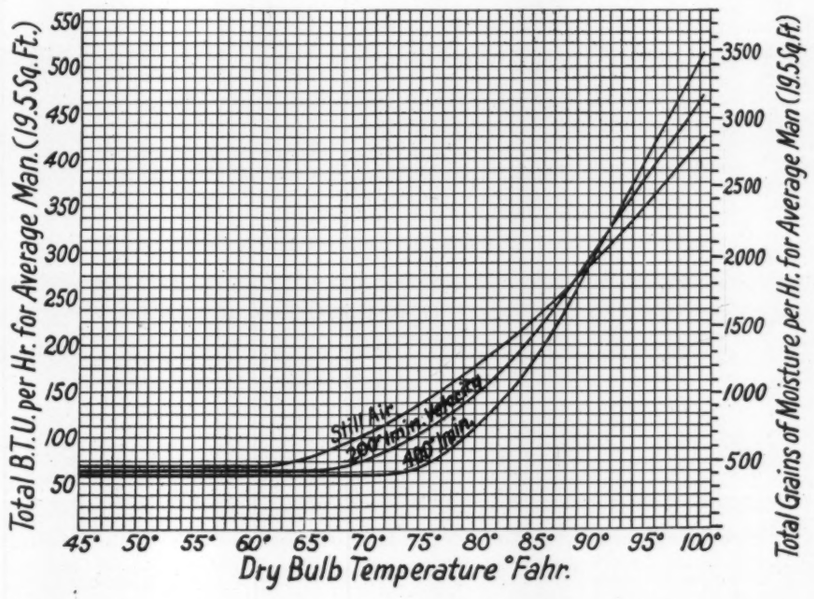


FIG. 8. RELATION BETWEEN HEAT AND WEIGHT LOSS FROM THE HUMAN BODY BY EVAPORATION AND DRY BULB TEMPERATURE FOR STILL AND MOVING AIR

Air conditioning in many industries is largely concerned with the control of the hygroscopic moisture. The moisture content of a hygroscopic material when in equilibrium depends upon the relative humidity of the surrounding air but varies widely for different materials.

The moisture content also varies to some extent for different temperatures at the same relative humidity. The content of hygroscopic moisture is sometimes termed *regain* and is expressed in parts of water per 100 parts of dry material. The curves (Fig. 4) show the relationship between moisture content and relative humidity for cotton and pure cellulose paper. These curves are given for temperatures of 70 deg. and 120 deg. fahr. Various investigators have given the regain of materials considerable study. Among the authorities for these various materials are Schloesing, Hartshorn, and the U. S. Forest Products Laboratory.

If the material contains moisture in excess of the regain corresponding to any relative humidity, then it will give up moisture to the air with absorption of heat and a resultant cooling effect equal to that of evaporation. On the other hand, if the hygroscopic material contains moisture below that of the corresponding atmosphere, it will absorb moisture from the air with a release of heat which corresponds to the latent heat given off if a corresponding quantity of water vapor were condensed.

Certain chemicals, such as a concentrated solution of calcium chloride, solutions of sulphuric acid, alcohol, etc., are correspondingly hygroscopic and react to changes in moisture in the same way. In certain cases, however, such as with sulphuric acid, there is not only the latent heat of absorption but also a chemical heat of combination.

Air Conditioning and Human Comfort

It has been shown that air conditioning has an important effect on the manufacture and processing of the products in many industries and at the present time its effect on human comfort, and hence on human efficiency, is becoming increasingly appreciated. In the cinema theatres and in the studios producing talking pictures human comfort is a highly important factor.

The advantages of air conditioning installations for

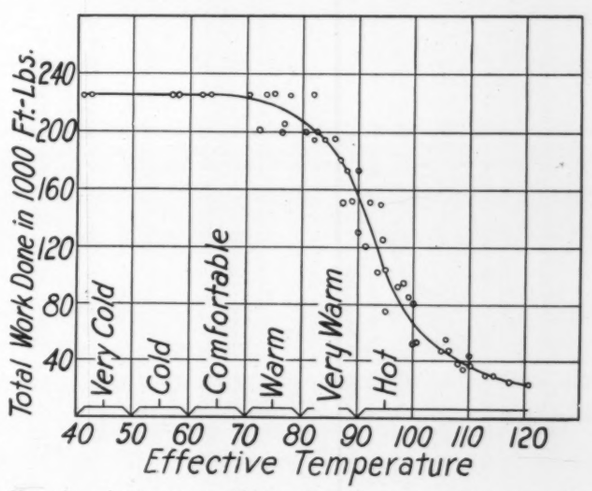


FIG. 9. AMOUNT OF WORK PERFORMED IN RELATION TO EFFECTIVE TEMPERATURE

human comfort are now being particularly stressed in many industries during the summer where the efficiency of operatives would otherwise be generally lowered. Likewise air conditioning is offering stimulus to the summer attendance at theatres and other places of entertainment.

Attention is called here to the important research engaged in by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service at Pittsburgh, Pa. Research in this field has been carried on continuously for the last eight years and the results have been published in the annual transactions of the Society.

Very valuable work on other phases of the relation of air conditions to human comfort has also been done under the direction of Philip Drinker, of the Harvard School of Public Health, Boston, Mass.

The subjects studied in these research programs may be classified under three separate heads.

1. The relationships of the three variables, temperature, humidity, and air motion, to human comfort and the quantitative relationship of these variables one to the other.
2. The determination of zones of optimum comfort embodying these three variables in winter and in summer.
3. Determination of the relation of sensible and latent heat given off by the human body for various combinations of temperature, humidity, and air motion so that this quantitative data required for the design of ventilating equipment for human comfort in audience halls and theatres might be available.

Attention is also called to the work of Dr. Leonard Hill of England who invented the kata-thermometer, an instrument which indicates the combined cooling power of the air and surrounding radiating surfaces under various conditions.

Effective Temperatures

It was determined by the investigations of the Research Laboratory of the A. S. H. V. E.[10] that in still air there is a definite mathematical relationship between wet and dry bulb temperatures giving conditions of equal comfort. That is, starting with saturated air at any given temperatures where the wet and dry bulb temperatures are identical, it was found that the same sensation of warmth could be obtained by increasing the dry bulb and proportionately decreasing the wet bulb temperatures.

[10] Determining Equal Comfort Lines. F. C. Houghten and C. P. Yaglou, TRANS. A. S. H. & V. E. Vol. 29, 1923.

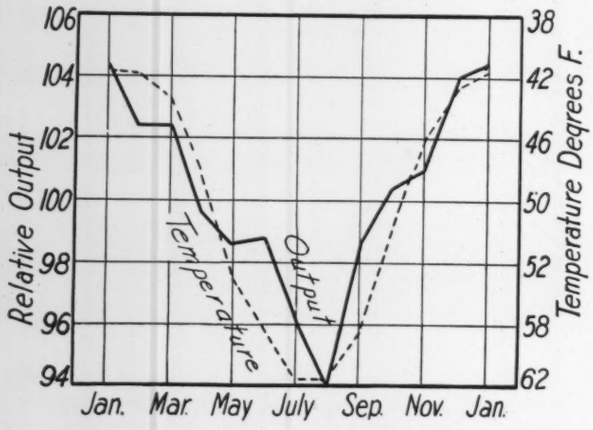


FIG. 10. AVERAGE RELATIVE OUTPUT IN FIVE TINPLATE MILLS OVER A PERIOD OF SEVEN YEARS

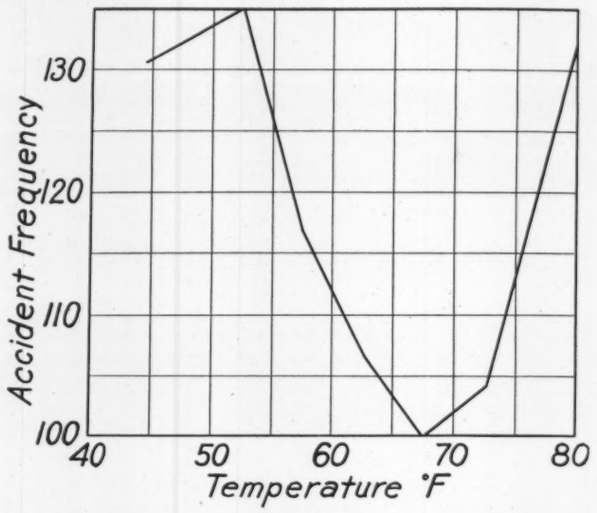


FIG. 11. FREQUENCY OF ACCIDENTS IN MUNITION FACTORIES ACCORDING TO OSBORN AND VERNON

Such a series of atmospheric conditions giving the same feeling of warmth when plotted on a psychrometric chart, such as given in Fig. 2, gives a straight line called an *equal comfort line*. A series of such equal comfort lines are superimposed on the psychrometric chart (Fig. 5). This series of equal comfort lines gives an index of a person's sense of warmth which is called the *effective temperature* scale. The effective temperature of any point on the chart is fixed by the dry bulb temperature of the intersection of the effective temperature line through that point with the saturation curve.

The slope of the effective temperature lines were found to vary with the degree of clothing worn by, and the activity of, a person. The effective temperature scale given in Fig. 5 is for persons normally clothed and slightly active in still air. Similar charts for persons with various degrees of clothing and work in still and moving air have been prepared.[11] In general then the effect of wearing less clothing is to increase the proportionate effect of wet bulb temperature or to swing the lines about their intersection with the saturation curve so as to make them more nearly parallel to the wet bulb temperature lines.

Air velocity tends to move the effective temperature lines to the right and decrease their angle with the dry bulb temperature lines so that a higher dry and wet bulb temperature is required for the same feeling of warmth with moving air. In all cases, the effective temperature is measured by referring it to the temperature of saturated still air which forms the basis of the effective temperature scale.

The scale of effective temperature is the only index

[11a] Cooling Effect on Human Beings by Various Velocities. F. C. Houghten and C. P. Yaglou, TRANS. A. S. H. & V. E. Vol. 30, 1924.

[b] Effective Temperature with Clothing. C. P. Yaglou and W. Edw. Miller, TRANS. A. S. H. & V. E. Vol. 31, 1925.

[c] Effective Temperature for Persons Lightly Clothed and Working in Still Air. F. C. Houghton, W. W. Teague and W. E. Miller, TRANS. A. S. H. & V. E. Vol 32, 1926.

[d] Application of Temperature, Humidity, and Air Motion Data to Air Conditioning Problems F. C. Houghten, W. W. Teague, W. E. Miller, TRANS. A. S. H. & V. E. Vol. 33, 1927.

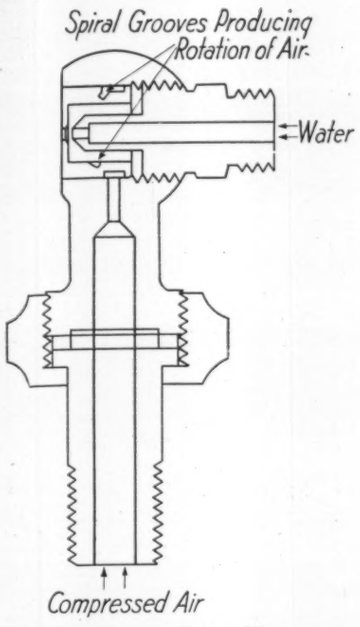


FIG. 12. DIRECT TYPE HUMIDIFIER HEAD

which truly indicates relative human comfort for all conditions of temperature, humidity, and air motion. The dry bulb thermometer alone has little value, contrary to the usual impression. If the humidity is low we must raise the temperature, or if the humidity is high we must lower the temperature, so that for any given effective temperature, many combinations of the dry bulb temperature and humidity are possible.

Research in order to determine the zones of human comfort and optimum comfort conditions for occupied space in winter was conducted by the Research Laboratory[12] of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Recently research at the Harvard School of Public Health[13] has shown that the effective temperature for optimum comfort in winter and summer differed considerably. Optimum comfort requires a higher effective temperature in summer than in winter, due, apparently, to acclimatization and to the difference in clothing worn. The optimum comfort lines for winter and summer as determined in Pittsburgh and Boston respectively are given in the chart (Fig. 6).

These researches not only indicate the conditions which should be produced by air conditioning systems in procuring human comfort, but also emphasize the necessity of moisture removal from the air on humid days as well as the necessity of lowering the temperature on hot days.

While the chart indicates that the temperature sensation of cooler saturated air may be exactly equivalent to that of dry air at higher temperatures, nevertheless there are other considerations besides the sensation of temperature that are important and should be taken into account. Dampness of the clothing and skin are undesirable, hence it is advisable to secure comfort in summer by using a lower humidity with a temperature higher than would be required with excessively moist air. In winter, on the other hand, when the moisture content is exceedingly low, comfort is better secured by providing higher humidity than outside conditions would normally give with correspondingly lower temperatures. In this way, the production of dust by electrification, the drying of furniture and the unnatural drying of the membranes of the nose and throat produced by dry indoor conditions normally prevailing in winter are avoided.

[12] Determination of the Comfort Zone with Further Verification of Temperatures within this Zone. F. C. Houghten and C. P. Yaglou, TRANS. A. S. H. & V. E. Vol. 29, 1923.

[13] The Summer Comfort Zone: Climate and Clothing C. P. Yaglou and Philip Drinker, TRANS. A. S. H. & V. E. Vol. 35, 1929.

Humidity and Air Movement in Relation to Comfort and Health

For human comfort, the range of satisfactory and practicable humidities is between 35 per cent and 65 per cent, the latter representing a maximum summer condition and the former a minimum winter condition.

The brilliant work of Huntington[14] indicates that a fairly high degree of moisture in the air is the most favorable. According to his investigations in the dry interior of the United States, a relative humidity of less than 30 per cent increases the death rate by at least 10 per cent above that which prevails when the relative humidity is 70 per cent or higher. He points out that in winter, when outdoor air is heated from about the freezing point to about 70 deg. fahr., an indoor climate as dry as that of the driest part of the continental interior is produced. Such dryness there in summer causes an increase of at least 10 per cent in the death rate, as shown by his climographic tables and charts.

Apart from its effect upon health, humidity, according to Huntington,[15] exerts an appreciable influence upon the working capacity of men. For the same temperature, he finds that neither the dry nor the wet days are favorable to the performance of physical labor, but that the best results are obtained with an indoor relative humidity of about 60 per cent at temperatures between 65 deg. and 70 deg. fahr.

[14] Ellsworth Huntington: *World Power and Evolution.* Yale University Press, 1919.

[15] Ellsworth Huntington: *Civilization and Climate.* Yale University Press, 1919.

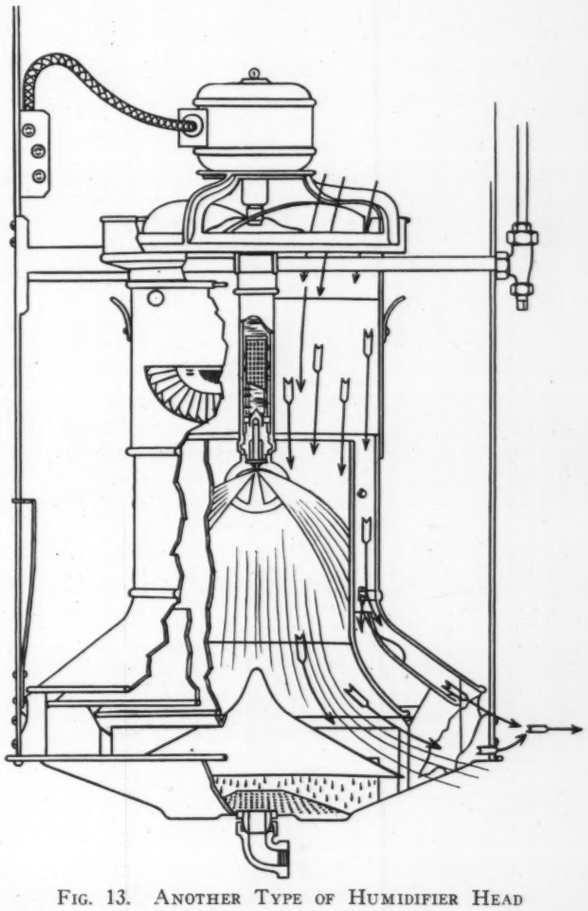


FIG. 13. ANOTHER TYPE OF HUMIDIFIER HEAD

Air Movement and Variability of Air Conditions in Relation to Comfort and Health

Air movement plays an important role in ventilation, not only because it blows away the warm and humid aerial blanket which surrounds the body, but also because it exerts a pleasant and stimulating effect upon the skin, a sensation which is lacking in still air.

According to Baetjer,[16] who studied the influence of air currents on comfort, "currents of air in the threshold of perceptibility contribute to our comfort in all probability by acting on the sensory nerves of the skin, directly or indirectly, and arousing sensations of an agreeable kind." Under ordinary room temperatures this investigation found that the threshold of perceptibility of air currents on the exposed surfaces of the body average in the neighborhood of 4.5 ft. per minute.

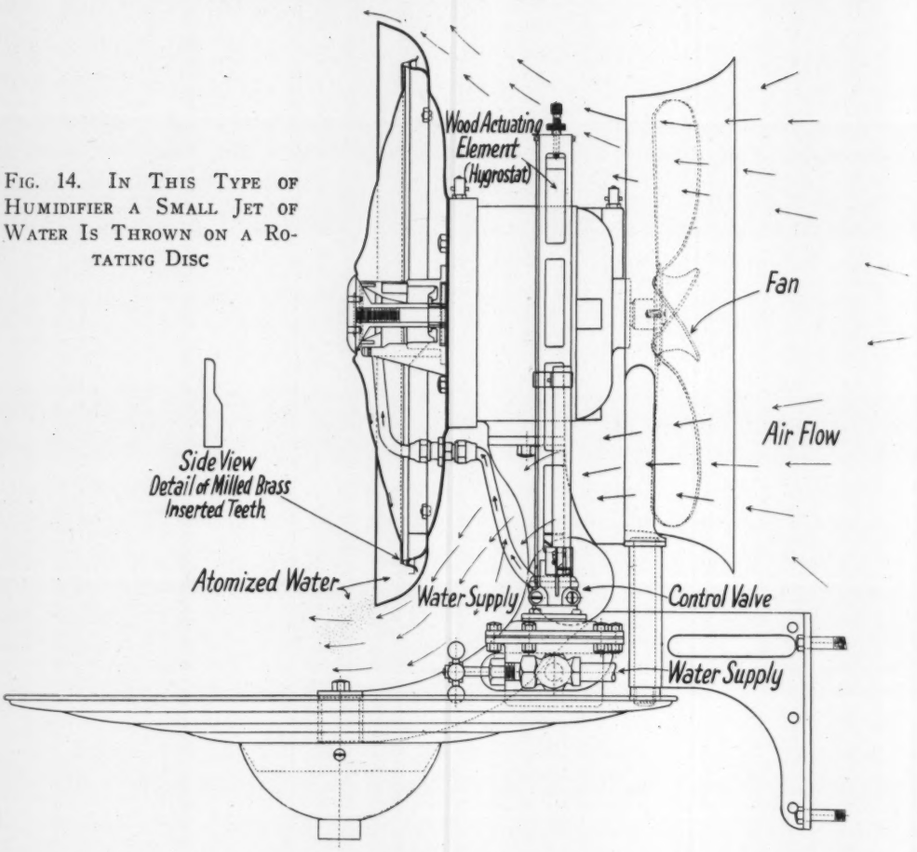


FIG. 14. IN THIS TYPE OF HUMIDIFIER A SMALL JET OF WATER IS THROWN ON A ROTATING DISC

Heat and Moisture Loss from the Human Body

Many experiments have been conducted for the purpose of determining the heat evolved by the human body, but the first work which shows the variation in the relation of sensible heat and the latent heat of evaporation to different air conditions was probably done at the Research Laboratory of the A. S. H. V. E.,[17] almost simultaneously with the work of Campbell and Angus[18] which was carried on in England.

The first of these researches is more thorough than the second and it is applicable to American conditions. It shows that the total heat emitted from the human body at rest varies but slightly under ordinary indoor conditions, averaging about 400 B.t.u. per hour for adult men. It is found, however, that the amount of heat lost by radiation and convection decreases rapidly as the dry bulb temperature rises, while that lost by evaporation is correspondingly increased. The relationship between

[16] Baetjer, A. M. Threshold Air Currents in Ventilation, *Amer. Jour. of Hyg.*, vol. IV, No. 6, Nov., 1924.

[17] Heat and Moisture Loss from the Human Body and its Relation to Air Conditioning Problems. F. C. Houghten, W. E. Miller, W. W. Teague, W. P. Yant, JOURNAL, A. S. H. & V. E., 1929.

[18] Physiological Reactions of Resting Subjects to Cooling Power and Effective Temperature J. Argyll and T. C. Angus. *Jour. Ind. Hyg.*, Vol. X, No. 10, Dec., 1928.

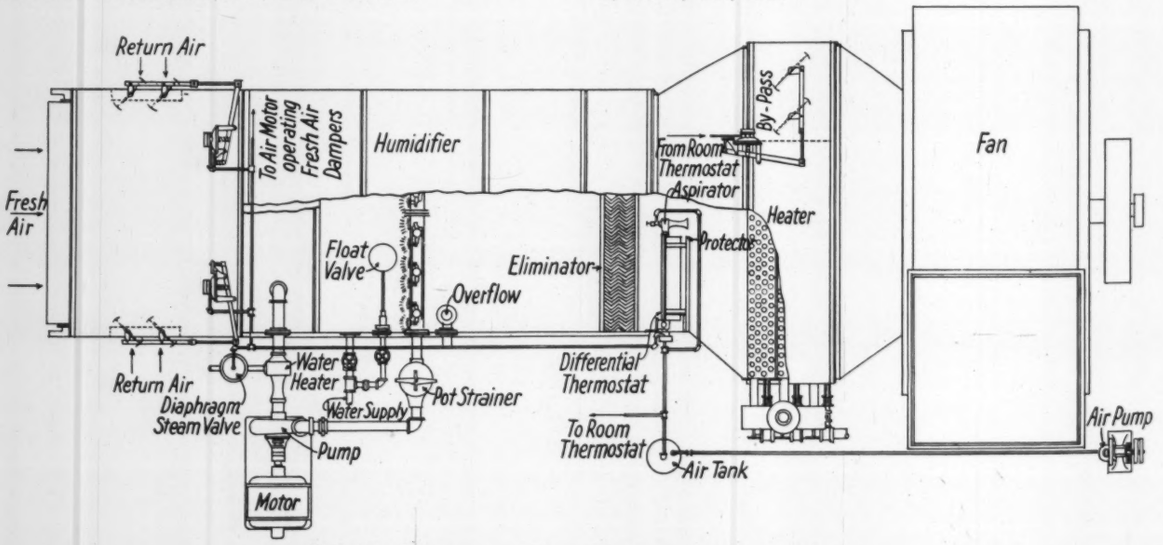


FIG. 15. HUMIDIFIER WITH HUMIDITY CONTROL—PLAN

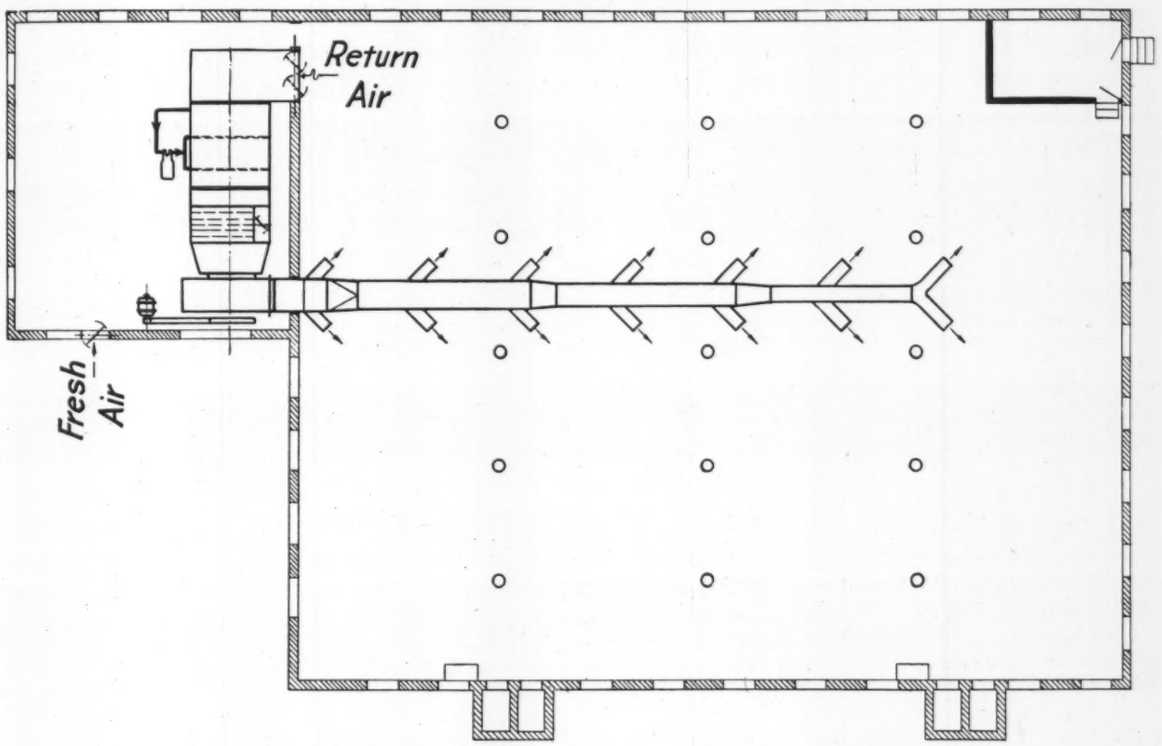


FIG. 16. GENERAL DESIGN OF ONE SYSTEM OF HUMIDIFICATION AND AIR CONDITIONING

sensible and latent heat losses for different dry bulb temperatures is shown in Fig. 7a and Fig. 7b. These afford valuable data for the design of ventilating and cooling equipment for human comfort. The variation of the amount of latent heat and evaporation at different air velocities is shown in Fig. 8.

Neither of these two groups of workers attempted to separate the purely radiation loss from the loss by convection. Important work of this nature is now in progress at the Smithsonian Institution and published preliminary data[19] indicate that the heat loss from the human body by convection is roughly one-third less than that lost by radiation, in still air of ordinary room temperatures.

The Influence of Air Conditions Upon Physical Work and Industrial Output

Laboratory data secured in various industries are all in accord in that atmospheric conditions exert a powerful influence upon physical activity.

In heavy physical labor, such as that involved in lifting dumb-bells or riding a stationary bicycle, the New York State Commission of Ventilation[20] found that the amount of work accomplished was 15 per cent less at 75 deg. fahr. with 50 per cent relative humidity than at 68 deg. with the same humidity, and 28 per cent less at 86 deg. with 80 per cent relative humidity. Vitiated air (23 to 66 parts of carbon dioxide per 10,000 parts of air) appeared to exert a distinctly harmful influence. Temperature and humidity remaining the same, the subjects performed 9 per cent less work in the stale air than in the pure air.

More recent and thorough data have been secured at the Research Laboratory of the A. S. H. V. E.[21] in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service.

It will be seen from Fig. 9 that the maximum amount of work was performed between the effective temperature limits of 40 deg. and 75 deg., although the sensations of comfort varied considerably.

In actual practice, Vernon[22] has shown that the output in British factories undergoes a distinct seasonal variation which varies inversely as the mean monthly outdoor temperature. Unfortunately, he did not record the actual air conditions in the factories. Fig. 10 shows the average relative output in five tinplate mills over a period of seven years.

More striking data are presented by Vernon[23] for British coal mines where the conditions are more severe than in the tinplate industry.

The relation of working capacity in the coal mines as determined by Vernon is shown in the following table:

Effective Temperature, Deg. Fahr.	Wet Kata Cooling Power	Rest Pauses. Minutes Per Hour	Tub Filling Time, Minutes	Relative Output, Per Cent
65.8	18.6	7.3	8.0	100
65.8	14.6	6.7	8.6	94
75.3	12.9	9.0	8.5	91
75.8	10.8	10.0	9.2	82
77.8	9.0	11.1	9.1	81
81.8	6.4	22.4	9.6	59

[19] A Study of Body Radiation. Smithsonian Miscellaneous Collections, Vol. 81, No. 6, Publication 2980, Dec., 1928.

[20] Ventilation Report of the New York Commission of Ventilation, E. P. Dutton & Co., New York, 1923.

[21] Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air. W. J. McConnell and C. P. Yaglou, TRANS. A. S. H. & V. E., Vol. 31, 1925.

[22] The Influence of Hours of Work and Ventilation on Output in Tinplate Manufacture. H. N. Vernon, *Industrial Fatigue Research Board, Report No.* 1, 1919, London, England.

[23] The Relation of Atmospheric Conditions to the Working Capacity and the Accident Rate of Coal Miners, Ibid, *Report No.* 39, 1927.

Other data very similar to these have been obtained by Orenstein and Ireland[24] in the Rand Mines in South Africa.

The Influence of Air Conditions Upon Industrial Accidents, Sickness and Mortality

Atmospheric conditions affect also the frequency of industrial accidents. The chart (Fig. 11) by Osborn and Vernon[25] shows that the frequency of accidents in British munition factories reach a minimum at 67.5 deg. dry bulb temperature, and that they increase above and below this temperature.

Under more adverse conditions, such as those met at the Moro Velho Gold Mine in Brazil,[26] twenty fatal accidents occurred in 16 months prior to the installation of a refrigerating and ventilating equipment, when the conditions underground averaged 101.5 deg. fahr. dry bulb, 87 deg. fahr. wet bulb, and 7.7 wet kata cooling power. During the 16 months after the installation of the equipment, only six accidents occurred, because the dry bulb temperature was reduced to 97.4 deg. fahr., the wet bulb was lowered to 76.2 deg. fahr., and the wet kata cooling power was increased to 20.5 milic. per square centimeter per second. Besides, the output of the workers increased by 12 per cent after the improvements were made.

Air Conditioning Equipment

Air conditioning equipment comprises:

1. Apparatus for providing the necessary heat and moisture exchanges within the room to be conditioned and of the air supplied to the room for ventilation.
2. Automatic devices for governing such apparatus so that temperature and humidity may be controlled within the limits desired.

The latter are referred to as controls. Those for controlling temperature are known as thermostats, and those for controlling humidity are known variously as hygrostats, psychrostats, or humidostats.

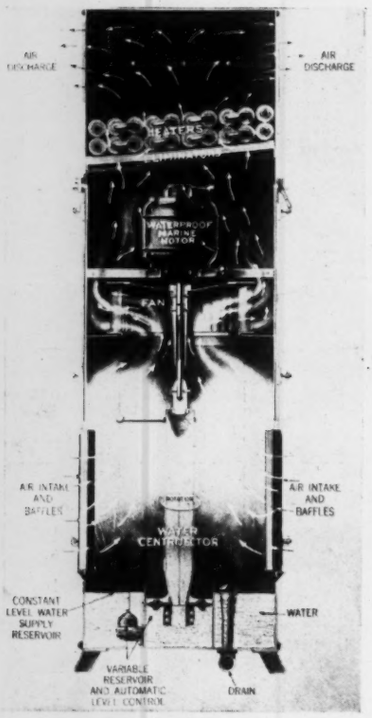


FIG. 17A. PHANTOM VIEW OF INTERIOR OF UNIT CONDITIONER

Apparatus for effecting humidity changes may be divided into two distinct groups:

1. The humidifiers used for the purpose of increasing the moisture content of the air and to produce cooling by evaporation.
2. The dehumidifiers used for the removal of moisture from the air and to produce cooling by contact with water or surfaces at a lower temperature than the air.

[24] A Contribution to the Study of the Influence of Mine Atmospheric Conditions on Fatigue. Orenstein and H. J. Ireland. *Jour. South African Institution of Engineers*, March 1921.

[25] Sixth Annual Report of the Industrial Fatigue Research Board, 1925, London, England.

[26] The Air Cooling Plant of the St. John del Rey Mining Co., Ltd., Brazil. E. Davies. *Trans. Inst. Mining Engrs.* Vol. LXIII, 1922, London, England.

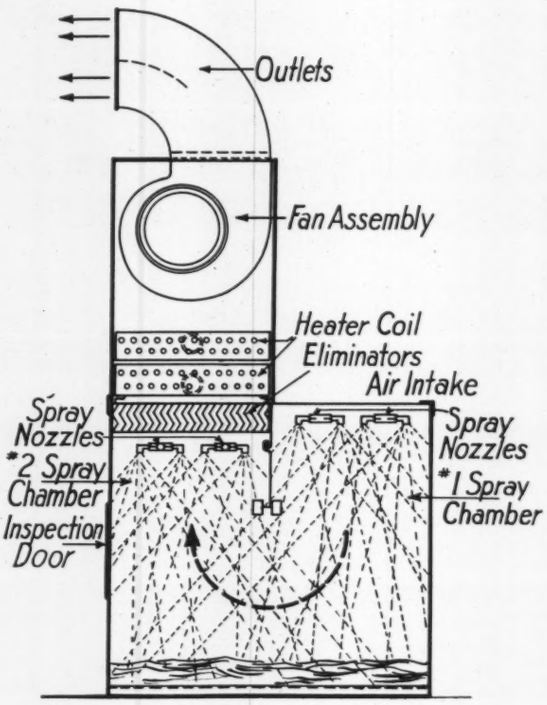


FIG. 17B. SECTION OF A UNIT CONDITIONER

Systems of Humidification

Humidifiers may be divided into the following general types according to the method of operation:

1. Indirect (Introduction of moistened air).
2. Direct (Spraying into the room).
3. Combined (Direct and indirect).

Before the advent of modern air conditioning the direct type humidifier was largely used for increasing the humidity in cotton mills and similar places. There has been much improvement in this type of humidifier and they still find wide application, especially in the textile industry.

One of these types uses atomizer heads operated by compressed air in which the water is drawn into the nozzle and then finally atomized by a small jet of air under high pressure. This type is illustrated in Fig. 12.

In another type, water is supplied at high pressure (usually about 200 lb. per square inch) to an atomizing nozzle. This nozzle is located in a sheet metal enclosure suspended from the ceiling through which a current of air is caused to pass, either by the action of a motor driven disc fan or by the induction action of the spray itself. The heavier particles of water are separated by a simple method of baffling and only the very finest particles are caused to pass into the room with the current of air where they are rapidly evaporated. This type is shown in Fig. 13.

In a third type of direct humidifying head a small jet of water is thrown upon a disc rotating at high velocity within a stationary circular comb which consists of fine metal strips or teeth against which the water thrown from the disc impinges. The finely atomized particles that are thus provided are carried out with the current

of air created by a disc fan operated from the motor shaft. This type is illustrated in Fig. 14.

The three types of direct humidifiers described are suitable only for application in certain types of industrial plants and do not provide either for controlled ventilation or for dehumidification, which is essential in many industries as well as for applications providing for human comfort.

Indirect humidifiers are similar in operation to the spray type air washers except that the water is sprayed directly against the incoming air. Such humidifiers comprise a chamber, usually from 6 to 8 ft. in length, through which the air is drawn at a velocity from 500 to 700 ft. per minute. Inside the chamber are placed one or more banks of spray nozzles distributed uniformly over the cross sectional area of the chamber. These nozzles create a finely divided spray through centrifugal action and require water pressures for effective humidification of from 35 to 45 lb. per square inch. At the intake of the humidifying chamber a set of baffles distributes the air and prevents the spray from escaping from the chamber, while at the outlet of the humidifier chamber an eliminator is provided. This eliminator, consisting of a series of metal baffles, is so designed as to separate all of the free, unevaporated moisture from the humidified air. Thus, the air leaving the humidifier is completely saturated, but without any entrainment or unevaporated water particles. The general construction of this type of humidifier is shown in Fig. 15. Such humidifiers are used only in connection with ventilating systems. They are practically always placed on the inlet side of the ventilating fan. Provision is made for warming the water in winter so that the air may be saturated at a

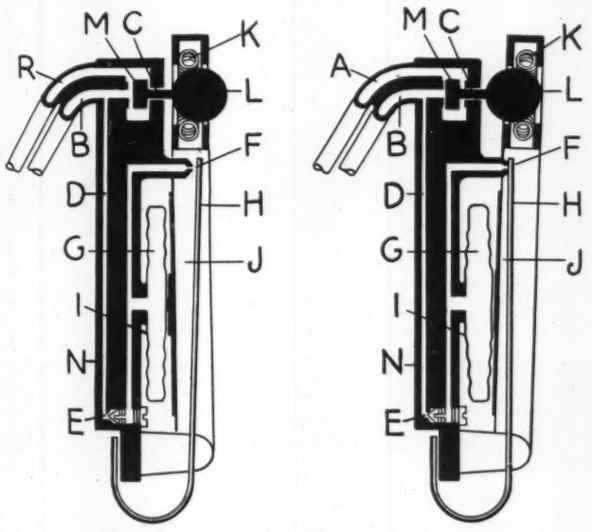


Fig. 18. A Bi-Metal Thermostat

definite temperature. The heating of the water is usually controlled by means of a thermostat placed in the path of the saturated air leaving the humidifier and controlling the heater through which the spray water passes.

In other cases, particularly in industrial work where high humidities are maintained, the control of the dew-point temperature is effected by a thermostat controlling the mixture of outside air and return air from the building by means of fresh and return air dampers at the inlet of the humidifier.

In industrial work, the dew-point regulation is effected in winter time by both of these means operating in conjunction. Where all outside air must be employed, it is usual to provide in front of the air washer, certain pre-heater coils of sufficient capacity to bring the air to a temperature slightly above the freezing point. In summer time operation, whenever the outside wet bulb temperature is above the minimum dew-point desired in the building, all outside air is taken and, it will be remembered from the psychrometric principles previously discussed, that the air issuing from the humidifier will have been cooled to the wet bulb temperature of the entering air, provided, of course, that the water is neither heated nor cooled by re-circulation, which is usually the case in such installations.

In most manufacturing processes relatively little moisture is added to, or extracted from, the air in the room by the process itself and, as windows and doors are closed, conditions in the room rapidly stabilize to the moisture content of the entering air. Thus, it is possible to control the relative humidity within the room by proportioning the quantity of the air in accordance with the heat sources within the room, such as heat from machinery and the occupants, electric lights, sunlight and outside radiation. The quantity of air to be supplied for each unit of heat thus removed is definitely dependent upon the relative humidity to be carried. For example, if 50 per cent relative humidity is to be carried in the room, the ventilation must be restricted so there shall be a rise of 20 deg. between the entering air and the air in the room itself. Smaller rises would give increased relative humidities. In textile mills, for instance, with the humidities usually carried, it is possible to keep the room temperature within about 12 deg. of the outside wet bulb temperature. Inasmuch as the wet bulb temperature on hot days is usually 15 deg. to 25 deg. lower than the dry bulb temperature, the mill temperature will be below the outside temperature and provision has to be made for the heat absorbed by inward transfer of heat through the walls. The general design of such a system of humidification and air conditioning is shown in Fig. 16.

The type of humidity control described is usually known as the *dew-point* control and is particularly applicable where there are a number of different departments to be served from one central equipment, and where there is but little change in the moisture content of the air in the room due to moisture changes in the materials processed or to excessive air leakage from outside.

Another system that is used considerably, especially where there is but one room to be controlled from one apparatus and where there may be considerable moisture change in the room itself due to local conditions, is to vary the dew-point at the apparatus, either continuously or intermittently, by means of a hygrostat placed in the room and so connected as to vary either the temperature or quantity of spray water used in the humidifier, or both. This system may be designated as the *direct relative humidity* control.

In such installation the temperature and humidity are both under control during the heating season, and in the summer time the humidity must be maintained constant but the temperature must be allowed to rise in accordance with the increase of the wet bulb temperature above the minimum desired. Temperature variations under

mill conditions will be from 75 deg. to 90 deg. fahr. Such results are much superior from the standpoint of comfort and well-being of the operators than those previously experienced. This has resulted in greatly increasing the effectiveness of the workers during this period, but the amount of ventilation provided has also greatly improved conditions in mills as to dust, lint, and odors. In the spinning of fine cotton yarns, this is of great value in producing an improved product as well as in bettering working conditions.

The uniform degree of air motion results in the lowering of the effective temperature with a consequent increase in comfort and efficiency of the workers at temperatures and humidities which are necessarily high because of the requirements of the product. The motion results also in removing the heat directly and immediately from the point of production on the spinning frame thus avoiding localized increase of temperature and decrease of relative humidity. In other words, it permits the entire process to be subjected more nearly to the average room conditions.

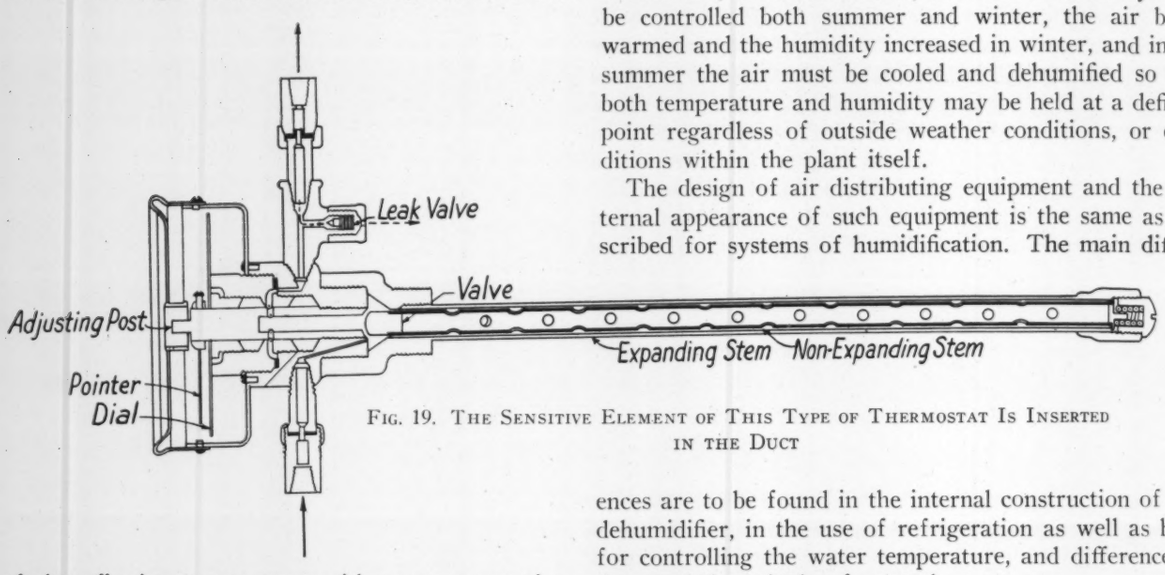


FIG. 19. THE SENSITIVE ELEMENT OF THIS TYPE OF THERMOSTAT IS INSERTED IN THE DUCT

Where large quantities of power are generated in a limited space, and where higher than 60 per cent relative humidity is required it is feasible and economical to use a combination of direct and indirect humidification. The indirect humidification is for the purpose of securing the desired quantity of ventilation and cooling, and the additional direct humidification placed in the rooms to increase the humidity and, at the same time, secure additional cooling effect. In general it may be stated that direct humidification is most satisfactory where higher humidities with but little cooling or ventilation are required.

Therefore, the indirect system is best where lower relative humidities with maximum cooling and ventilating effects are desired, while for conditions where there are both large quantities of heat to be absorbed by ventilation and high humidities are needed the combination system of direct and indirect is more desirable.

Inasmuch as the cooling and humidification requirements in summer are much more severe than those in winter, an excess of humidifying and cooling capacity must be provided to meet extreme conditions and this requires, for the best results, careful automatic control. Such applications of humidity control may be considered standard practice today in the United States.

The same type of equipment as used in cotton mills is largely applicable to other textile industries, including rayon, and also to tobacco factories, paper mills, and other buildings.

Systems of Dehumidification

In many industries, as already pointed out, it is as important to control the temperature in summer as in winter and, at the same time, the relative humidity must be controlled both summer and winter, the air being warmed and the humidity increased in winter, and in the summer the air must be cooled and dehumified so that both temperature and humidity may be held at a definite point regardless of outside weather conditions, or conditions within the plant itself.

The design of air distributing equipment and the external appearance of such equipment is the same as described for systems of humidification. The main differences are to be found in the internal construction of the dehumidifier, in the use of refrigeration as well as heat for controlling the water temperature, and difference in the general methods of control.

Dehumidifiers are of two general types. *First,* the spray type in which the water is cooled outside of the spray chamber and then introduced. *Second,* the type in which the refrigerating coils are placed directly in the spray chamber and the water is sprayed over this surface, air coming in contact both with the wetted coils and the spray.

Both types of dehumidifiers have air distributing baffles at the inlet and eliminators at the outlet. The dehumidifier is usually considerably longer than the humidifier and in the spray type there are two sets of sprays, one or both of which is opposed to the air flow.

The individual spray nozzles are considerably larger in capacity than the spray nozzles used in the humidifier and operated at a much lower pressure, usually from 20 to 25 lb. per square inch, the object being to get a coarser spray and, at the same time, a much larger quantity of spray per unit of air to be treated than in the humidifier. An exceedingly finely divided, or atomized, spray is not as effective in dehumidification as there must be provided heat capacity as well as surface in the spray water. The usual allowable rise in spray water temperature entering and leaving the dehumidifier is from 6 deg. to 10 deg. fahr., and in well designed dehumidifiers of the spray type the final temperature of the air is substantially identical with the final temperature of the leaving water. This is made possible by the counter flow effect employed in such apparatus. The velocities through the dehumidifier are usually lower than those employed in the humidifier, normal velocities being from 400 to 550 ft. per minute under the cross sectional area.

Greater care must also be taken with the elimination

of free water than with the humidifier as an entrainment beyond the eliminator defeats the purpose of the equipment.

Dehumidifier systems are usually provided with two essential control elements, a dew-point control at the apparatus and a room control. The former maintains a uniform temperature of saturated air leaving the dehumidifier at all times of the year. In summer, in industrial work, this is so constructed as to use nearly all return air, thus reducing the refrigerating load required, while when the wet bulb temperature outside is at all

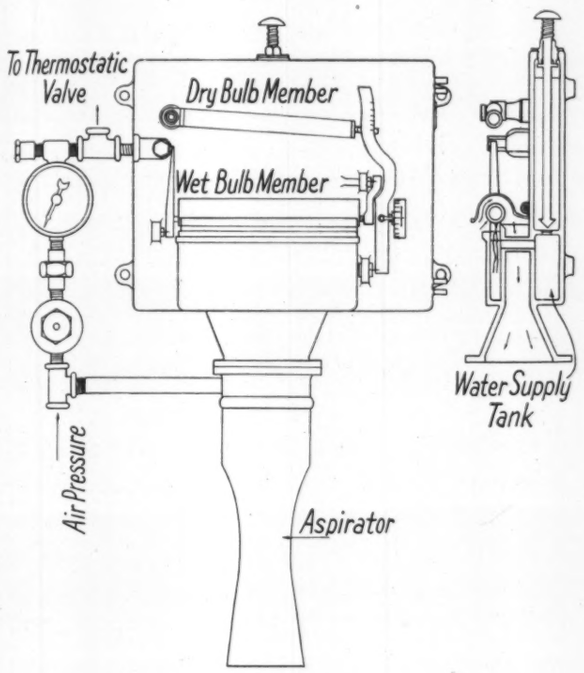


FIG. 20. DIFFERENTIAL HYGROSTAT

below the dew point required within the room the latter is controlled by a mixture of fresh and return air as required under control of the dew-point thermostat. By the second element the temperature in the room is independently controlled at a definite point above the dew-point or saturation temperature either by a thermostat or a hygrostat, depending upon whether the most important factor is the temperature or the relative humidity. Such control either operates upon volume dampers controlling the quantity of air introduced, or upon heaters which supply the necessary heat for temperature regulation. Usually the control is a combination of both volume control and the heating of the air.

In auditorium installations the requirements are quite different, since there must be a considerable volume of air circulated for ventilation and cooling purpose and for effective distribution. The moisture content must be relatively constant but its temperature varies according to the requirements of the auditorium. Satisfactory solution of this problem has proved difficult, but special systems of control are now in general use which permit the temperature and humidity to be controlled independently in the auditoriums and, at the same time, to be immediately responsive to changes in occupancy, and for the most part this is accomplished without the use of any external heat.

Systems of Air Distribution

With systems of air conditioning, the method of air distribution, as well as the practical details of design, are of utmost importance.

In an installation where exact relative humidity control is required uniform temperature conditions are very important because a variation in relative humidity will be caused by any variation in temperature. Moreover, the air is usually admitted, especially in summer, at temperatures lower than that of the room in order to effect the necessary cooling. Therefore, the air must be distributed uniformly, not with respect to floor space but with respect to the amount of heat to be absorbed in the various portions of the room. Thus it is necessary for the air conditioning engineer to have exact information as to the distribution and sources of heat within the room, or else to have means of controlling the temperature or quantity of air supplied to the different sections independently.

A great deal of attention has been given to various methods of air distribution and for industrial work adjustable diffuser outlets have been developed which will spread the air uniformly in a fan shaped manner. Such an outlet is also provided with a louvre damper to permit reduction of air velocity without changing distribution.

A second system of distribution utilizes a series of nozzles near the ceiling discharging horizontally at high velocity from one side of the room toward the other. The size of the nozzles used and their spacing depend upon the distance to which the air is to be carried. Such a system depends for its effect primarily upon the ejector action of a high velocity current of air which tends to produce a secondary current in the opposite direction over the area to be conditioned. The volume of the secondary air current when set into motion is usually several times that of the primary air discharged, the ratio depending upon the velocity of the air leaving the nozzles. This system is termed the ejector system of air distribution and is particularly effective in securing a thorough air mixture and uniformity of temperature and humidity. It has been applied successfully to the cooling and air conditioning of auditoriums as well as to certain industrial problems, particularly those involving the drying and processing of materials.

A *third* system of distribution aims at producing the required results in an entirely different manner, and is known as the *pan system* of air distribution. In this system the air is distributed entirely from overhead, one outlet being provided uniformly for each unit of ceiling area. The air is discharged downward against a flat plate and is thus diffused radially in all directions, laying a blanket of cool air which descends and mixes with the other air by gravity. This is to admit the air with the minimum effect of disturbance in the room itself and in installations for auditorium cooling it is effective in avoiding drafts. It is also very useful in certain industrial applications where cooling is required and it is desired not to disturb the heated areas, or to distribute dust or fumes throughout the building.

A *fourth* method, which was at one time most largely used, is that of distributing the air through mushroom outlets from the floor. In auditorium cooling, however, this has been objected to owing to the stratification of

air and to the directing of cold air toward the feet and ankles of the occupants.

Unit Air Conditioners

The indirect humidifiers and dehumidifiers just described are what are known as the central station type. That is, they consist of separate pieces of equipment, such as humidifiers, pumps, fans, distributing ducts, heaters, controls, etc., all assembled at the point of erection. They usually consist of but few sets of apparatus; more often only one set of equipment for the entire installation.

There is now being introduced quite extensively what is known as the unit type of air conditioner. (Fig. 17a and 17b.) With this type of equipment the requirements of a given room are met by installing the necessary number of standard complete air conditioning units, usually without any distributing duct work, the necessary air distribution being accomplished by the location of the units themselves. Each unit is a complete air conditioning plant in itself including spray air conditioners (humidifier and dehumidifier) air re-heater, fan, spray pump, automatic control, etc. These are usually built in graduated standard sizes having capacities from 2,500 to 10,000 cu. ft. each.

There are several advantages of such equipment as well as some disadvantages. The principal advantages are that the desirable results of the indirect humidification system can be applied to old buildings without installing duct work. Also where manufacturing processes are changed and departments are subdivided or enlarged the units are easily moved or increased in number to suit the new requirements. They are also particularly adapted for use by tenants of rented buildings since they are movable and do not become a part of the property when installed. Under certain conditions there is a more than sufficient saving by avoiding elaborate duct work to pay for the cost of the increased number of units. Standardized production and costs are also an added advantage.

On the other hand, air conditioning units, especially in manufactories where considerable dust and lint occur, require frequent attention and cleaning. Also some attention must be given the adjustment of controls. Such a multiple unit installation, therefore, often involves a great deal more attention than does a single unit located in the basement. In dehumidifying plants, especially, there is a great advantage in having a single air conditioning unit located conveniently to the refrigerating apparatus rather than to have a large number of water connections for supplies and returns. For dehumidification especially the central station system is to be recommended where more than three unit conditioners would be required.

Humidity Controlling Devices

Devices used in temperature control in air conditioning equipment are largely common forms which have long been in use and are well known. There are three general forms of these thermostats in use:

First, the most common, is the bi-metal thermostat which operates on the principle of the differential expansion of two metals having different coefficients; these metals being brazed together in a curved strip. The result of a change of temperature is to cause a change in the curvature of this strip thus actuating a contact arm, or lever, a pneumatic valve or an electric contact which provides the motive power for the actual operation of the controlling valves or dampers. (Fig. 18.)

The *second* type employs a metal diaphragm or bellows filled with a volatile liquid, the vapor pressure of which changes with change of temperature and actuates directly a pneumatic valve through which it controls the governing valves or dampers in the air conditioning apparatus.

The *third* type of thermostat is the duct thermostat in which the sensitive element is inserted in the duct, while the setting dial and air pressure connections are on the outside. This type of thermostat consists of an outside tubular thermo-responsive casing of brass or aluminum, or other highly thermo-responsive metal, and an inner rod or tube of special nickel steel or other metal, which is relatively unaffected by temperature changes. The valve is carried directly upon the inside non-expansive element. The outer expansion element and the inner element carries the pneumatic valve which operates directly upon an adjustable seat connected to the expansive element. This is also provided with a constant leak. The

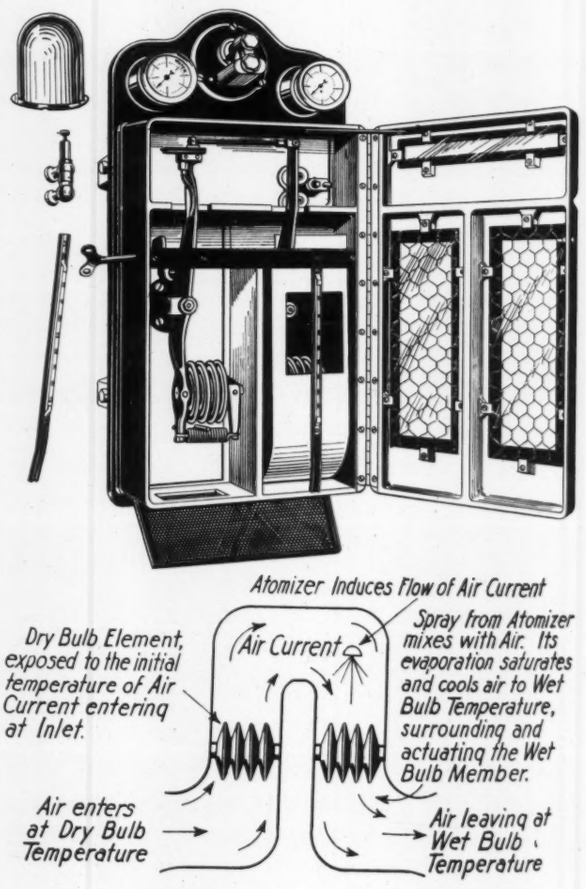

Atomizer Induces Flow of Air Current
Dry Bulb Element, exposed to the initial temperature of Air Current entering at Inlet.
Air Current
Spray from Atomizer mixes with Air. Its evaporation saturates and cools air to Wet Bulb Temperature, surrounding and actuating the Wet Bulb Member.
Air enters at Dry Bulb Temperature
Air leaving at Wet Bulb Temperature

FIG. 21. A & B. A COMMON FORM OF HYGROSTAT

action of this air supply valve in connection with the constant leak is such as to increase or decrease the pressure upon the operating diaphragms of the valve, or damper air motors. This type of thermostat is very rugged and responds to slight temperature changes with proportionate graduated effect. The construction of this type of thermostat is shown in Fig. 19.

Hygrostats are of two general types. The *first* type

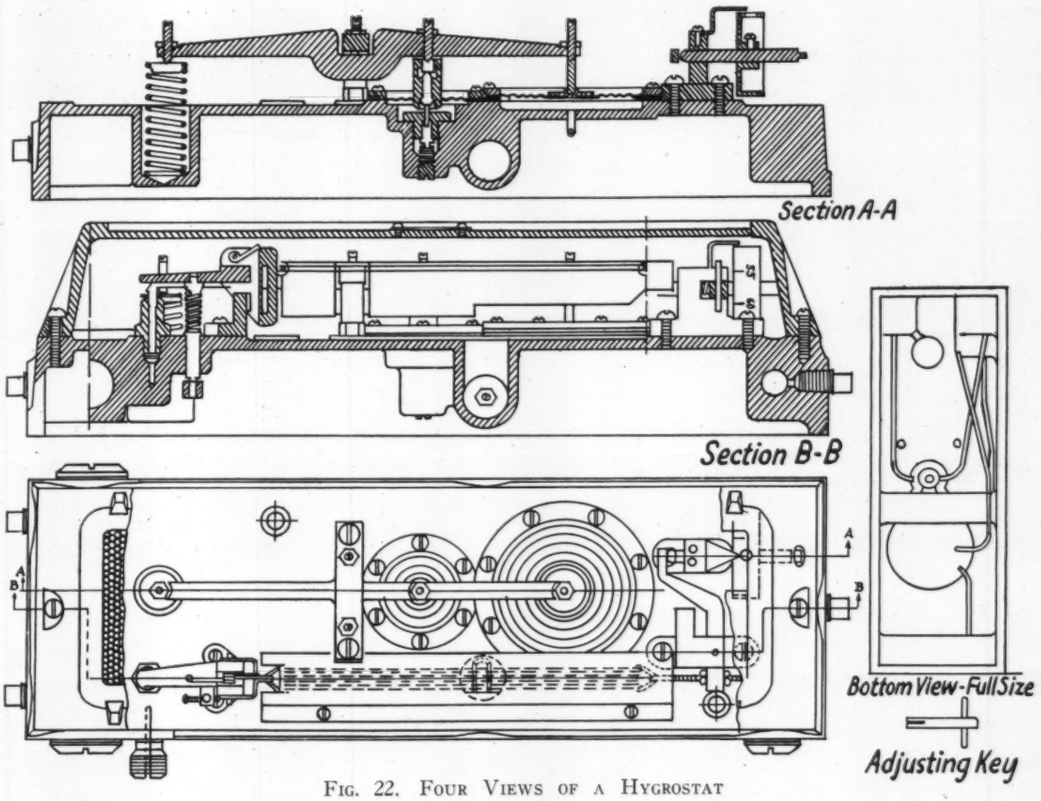


FIG. 22. FOUR VIEWS OF A HYGROSTAT

is in effect a form of differential thermostat in which one expansive element is affected by the wet bulb temperature, which temperature is suitably maintained either by a wick or a spray of water. These thermostatic elements are so inter-connected that they will not be affected by changes of temperature alone, but only by changes in the relation between the wet and dry bulb temperature which would result in changing the relative humidity. Such a correlation is made possible by the fact that there is practically a straight line relationship between the relative temperature changes of wet and dry bulb for any definite relative humidity (Fig. 20).[27] As in the thermostat, these elements may be either of the solid expansion or vapor pressure type. A common form of this apparatus is shown in Fig. 21 (a and b).

The *second* form of hygrostat is that depending upon the dimensional changes of hygroscopic materials due to changes in moisture content. In these instruments the humidity responsive element works directly upon pneumatic valves or electric contacts. The sensitive element responds to relative humidity and affects the actuating mechanism precisely as the thermostat responds to temperature.

It is found that suitable hygrostatic materials show no material change in extension with changes of temperature so long as the relative humidity is constant, although the actual moisture content of the hygroscopic material itself undergoes slight changes under these conditions. Certain types of these hygrostats have been made most sensitive and reliable. (See Fig. 22.)

Temperature and humidity controlling devices usually act upon the large control valves or dampers in the air conditioning apparatus through the medium of compressed air or electric currents. In either case the actuation is accomplished by suitable motors, either pneumatic or electric. For all the larger installations the pneumatic control is practically universally used on account of simplicity, reliability, and also on account of the fact that it permits graduation of dampers or valves with temperature or humidity changes. In the smaller humidifying devices electric controls are very serviceable because of simplicity of the circuits and from the fact that there are no auxiliaries, such as air compressors, air

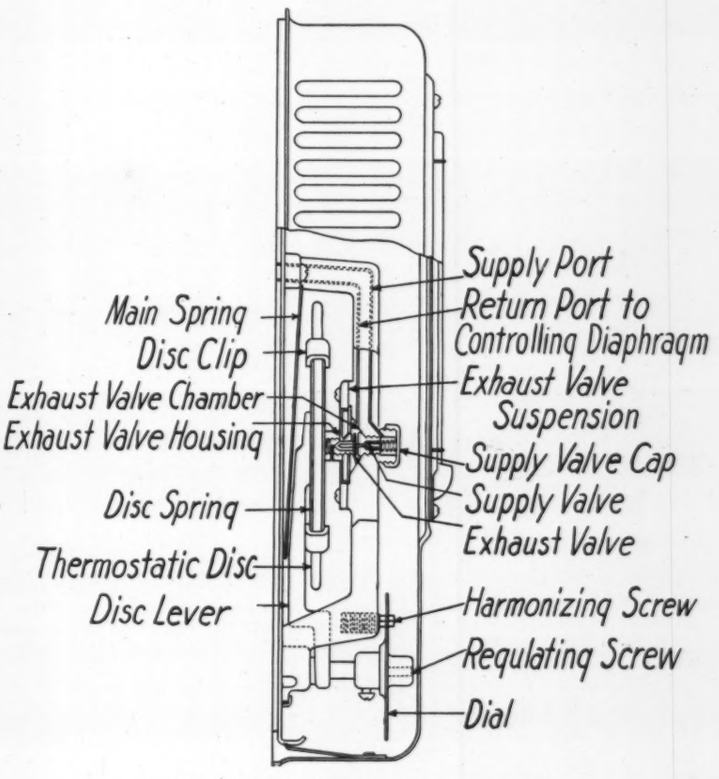


FIG. 23. APPLICATION OF THERMOSTATS

[27] Air Conditioning Apparatus. W. H. Carrier and F. L. Busey, *Trans. A. S. M. E.* Vol. 33, 1911.

tanks, air filters, relief valves, etc., required. In the larger systems such accessories are warranted by the increased size of equipment and the increased number of controls operated from the same compressed air source. In the unit types of equipment, direct acting thermostats and hygrostats are sometimes used in which the responsive element works directly upon the valves controlled without the intermediary use of either compressed air or electric current. Such applications are illustrated for thermostats in Fig. 23, and are shown for hygrostats in Fig. 14.

Summary

Modern air conditioning provides many devices for increasing or decreasing relative humidity within an enclosure. It also provides automatic means for the control of both temperature and humidity at any desired point regardless of external weather conditions and regardless of the effects of occupancy, or of manufacturing processes. It has resulted not only in great economic savings in many industries but has permitted standardizations of manufacturing which were heretofore impossible owing to variations in weather conditions. It has in many instances greatly improved the quality and uniformity of the product. Moreover, it has much improved working conditions in many industries and added greatly to the efficiency of the workers. It has afforded conditions of maximum comfort in crowded theatres and auditoriums during the hot summer months as well as in the winter, thus greatly increasing the return from the investment in such properties. In short, air conditioning has contributed both to the wealth and personal comfort of human society.

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Drunk on Bad Air

H. M. Johnson, head of the psychological laboratory of the Mellon Institute of Industrial Research, University of Pittsburgh, has been studying mental activity and work output of experimental subjects exposed to some degree of air pollution by means of exhaust gases. Members of the Simmons Fellowship for the Study of Sleep had made the observation that ordinary fatigue and the conditions of narcosis from drugs and from oxygen depletion depend alike upon asphyxiation of the muscle cells, and the tests were undertaken to discover in physiological and mental signs what degree of delayed response and impaired efficiency may be taken as an objective index of fatigue.

Throughout all tests the powers of concentration, steadiness, persistence, substitution, calculation, memory, and the ability to sustain continuous reaction ran such close parallels with increasing symptoms of asphyxiation that Mr. Johnson declares no distinction exists between so-called "nervous" fatigue and physical exhaustion. Almost the entire range of disturbances observed relates to factors in the physical environment, that is to say, to oxygen supply, air conditions, and comfort.

"A man gets drunk on too low an oxygen intake. He is at first pleased, buoyant, and willing to continue set tasks indefinitely along the same lines of endeavor that have brought about his inherent 'tiredness.'

"His attention narrows. He cannot be diverted from the thing that occupies him. He resents outside disturbance. He is unable to stop himself except by great endeavor.

"He becomes dull, clumsy, slow to respond, stupid. He says silly things that give himself away. He winks and blinks and grins, and gets emotional flares out of situations that are extremely silly.

"In advanced stages he begins to talk to himself. He converses with inanimate objects, reproaches a post or chair with tears or threats or curses. He may see double. His speech is thick. He is on a drunk, asphyxiated by the combustion products within his muscle cells.

"Three conditions counterfeit this picture of alcoholic poisoning: (1) a diminished oxygen supply, due to great dilution of the air by means of other gases, or in rarefied air of high altitudes, or in inclosed spaces polluted by carbon monoxide; (2) a lowered rate of cell respiration caused by alcohol or other drug; and (3) induced fatigue, when cells burn faster than the toxic products of incomplete combustion can be washed away by the blood stream."

Working to Advantage

The point is clear that any workplace demanding higher types of coordinated effort shall have fresh, abundant air; that better output and greater concentration of effort may hinge upon a slight fatigue; that the personal testimony of fatigued individuals cannot be relied upon to determine what air or other work conditions are favorable to good work and long continued maintenance of highest output; that the worker is a part of his own environment and, more particularly, the conditions of air environment cannot be studied apart from the worker himself and his characteristic reactions.

PRACTICAL PIPING PROBLEMS

Regulating Valve for Controlling Low Pressure Steam Tapping a Drain Opening

by **W. H. WILSON**

Steam Regulation

THE accompanying illustrations show a standard type of regulating valve, arranged for controlling low pressure steam. This valve is of the weight adjusted type, adjustment being made for the different pressures by moving the weight *D*. The steam inlet is shown at *A* and the outlet, or delivery side of the valve, at *B*.

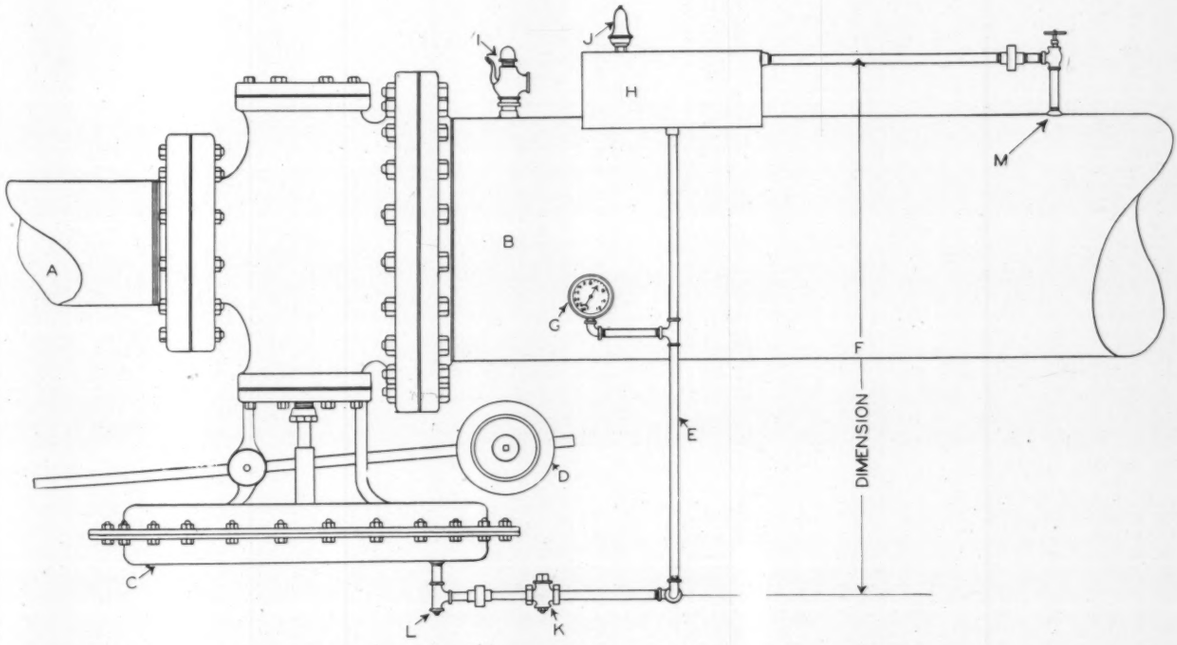
The illustrations, in detail, show the pressure-controlling pipe connection on the low-pressure pipe line and the positions of the flexible diaphragm, in the diaphragm chamber *C*, when the valve is in the open and also in the closed positions, it being understood that the reduced or delivery pressure acting on the lower side of the diaphragm with the aid of the valve stem moves an interior balanced valve disc in the body of the regulator.

In this type of regulator the diaphragm pressure must also overcome the resistance of the valve stem packing box. The expanded outlet type of valve which has the outlet or delivery side of the valve larger than the inlet side is frequently used when there is considerable difference between the inlet pressure and the outlet or service pressure.

The type of valve shown is controlled by the delivery or service pressure coming to the diaphragm through the small pressure control or governor pipe. Other types of valves have the diaphragm chambers connected integrally with the body of the valve, permitting the pressure on the service side of the valve to come in contact with the side of the diaphragm nearest the valve, which would be the upper side in this case. The small pressure control pipe connection is not required for a valve of this type.

By using a regulating valve with the controlling pressure under the diaphragm and the diaphragm chamber separated from the valve proper, the small pressure control pipe can be connected at any desired point, within a reasonable distance, where a steadier pressure exists. The small amount of labor that this connection requires will be well repaid in the better working of the valve.

A diaphragm chamber of this type has another desirable feature that frequently passes unnoticed but is a help to maintenance men. Should the rubber diaphragm break or "blow out," which is not uncommon, the steam pressure need not be shut off the regulator. All that is necessary is to shut off the small pressure control pipe and if the regulator has no by-pass connection, the lever is blocked in a position that will pass approximately the amount of steam that is being used while the diaphragm



STEAM REGULATION VALVE SHOWING PRESSURE CONTROL PIPE CONNECTIONS

is renewed. When the required service pressure is low, the diaphragm chamber should be extra large, to give more surface to provide the power for the low pressure to close the valve.

A diaphragm-operated valve is usually installed with the diaphragm chamber downward, but it may be installed with the diaphragm chamber at the top, in which case the pressure control piping should be arranged to provide water on the diaphragm at all times so that the steam will not come in contact with the rubber diaphragm.

When the initial pressure is high, in comparison to the delivery pressure, say 200 lb. initial pressure to ½ lb. delivery pressure, it is usually controlled by two valves, the first valve reducing to approximately 75 to 50 lb. and the second valve from the secondary pressure to the required low pressure.

Connecting the Control Pipe

The proper type of regulating valve having been selected for the requirements, the next thing of importance is to connect the pressure control pipe at the place where the pressure is steady and uniform in order that the regulator will work to its best advantage. The location of this connection should be carefully considered, as many of the troubles in connection with the operation of regulators and reducing valves can be traced to the pressure control pipe connection. When the outlet of the regulator is connected to a larger steam main, the pressure control pipe should be connected to the larger pipe line. If the pressure control pipe is connected to a steam main near the exhaust connections from large reciprocating engines or air compressors, the strong pulsations of the exhaust will be transmitted to the diaphragm and the varying pressure will be transmitted to the regulating valve. The pressure control pipe should be connected to a place where the pressure is steady, even if it happens to be some distance away. There is no objection to running this pipe a hundred feet distant from the valve, provided it is installed without air pockets. In this case, however, the pipe should be increased in size.

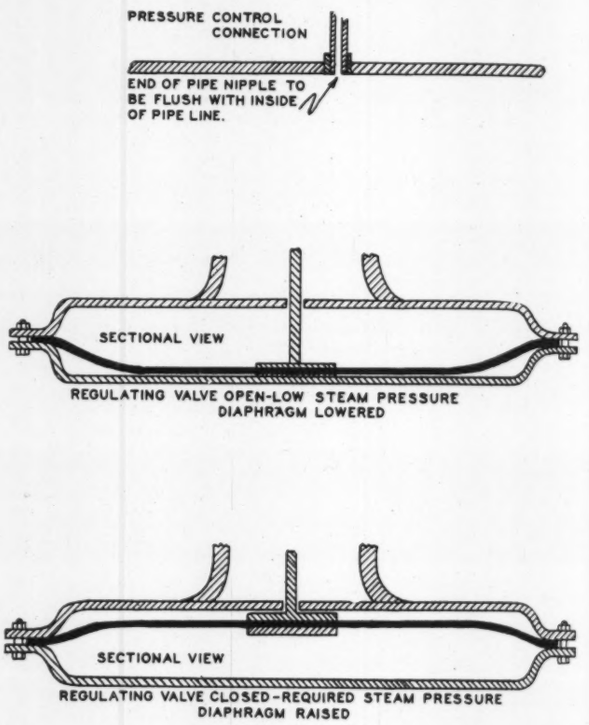
In case it is impossible to secure a place for the pressure control pipe that will be free from pulsations and provide a steady pressure, a large receiver should be used for the steam from the regulator to flow into and the receiver should connect to the steam main—the pressure control pipe for the regulator to be connected to the receiver.

The spasmodic or jumping action of the regulating valve, due to the pulsating and unsteady pressure through the pressure control pipe, can be checked somewhat by throttling the valve or stop cock that should be installed in every pressure control pipe (shown at *K* in the illustration). Install this stop cock near the diaphragm chamber of the regulator.

The Pipe Nipple

Particular attention should be given to the pipe nipple for the connection of the pressure control pipe to the steam main shown at *M*, and in the detail. This nipple should not extend beyond the inside surface of the pipe line as the flow of steam in the main pipe line may It is a simple matter when welding or installing this affect the pressure in the small pressure control pipe. nipple, especially in light weight pipe, to allow it to project through the wall of the pipe. It is better to take a little more time and see that it is inserted carefully and exactly flush with the inside of the pipe. The water reservoir shown at *H* should be applied to the pressure control pipe when accurate regulation at extremely low pressure is desired.

The pressure on the lower side of the diaphragm, when the regulator is in operation, will be the amount of pressure in the steam main *B* plus the weight of the


PRESSURE CONTROL CONNECTION
END OF PIPE NIPPLE TO BE FLUSH WITH INSIDE OF PIPE LINE.
SECTIONAL VIEW
REGULATING VALVE OPEN-LOW STEAM PRESSURE DIAPHRAGM LOWERED
SECTIONAL VIEW
REGULATING VALVE CLOSED-REQUIRED STEAM PRESSURE DIAPHRAGM RAISED

Interior Views of Diaphragm Chamber

column of water from the lower side of the diaphragm to its height in the vertical part of the control pipe *E* which will be to a point near the top of the reservoir.

Function of the Reservoir

If the reservoir were not in this pipe connection, when the diaphragm would raise with the increased pressure, as shown in detail, more water would flow in the diaphragm chamber, causing the height of water to drop in the vertical pipe before more could be added by condensation. The head of water would be reduced in the vertical pipe, with a corresponding reduction in pressure on the diaphragm, which in turn would cause the valve to be slower in shutting off, allowing the steam pressure to build up in the steam main *B*. The reverse action takes place when the valve begins to open.

The rise and fall of water in the vertical section of the control pipe, when the regulator is required to deliver steam at extremely low pressures, causes fluctuations in the governing pressures. This can be improved by installing a water reservoir. When the water reservoir is used, the raising of the diaphragm will cause water to flow in the diaphragm chamber as before, but instead of lowering the head of water in the vertical pressure control pipe the reservoir will supply the water, and on ac-

count of its large surface the vertical drop of the water will be very slight. The pressure head of water will be maintained at all movements of the diaphragm and smoother operation of the regulator and more uniform delivery pressure will result. The reservoir must be placed at the highest point in the pressure control pipe connection. A thermostatic air vent is placed, as shown at *J*, in case any air or other gases should, by chance, get into the reservoir.

The size of the reservoir depends on the capacity of the diaphragm chamber and the size of the pressure control pipe connection. Ordinarily, for a large regulator, a piece of 10- or 12-in. pipe, 6 in. long with pieces of flat steel welded in for heads, can be easily and quickly made in the shop and will be suitable; for smaller diaphragm chambers, these sizes may be reduced. The height of the reservoir is not important, as it is the surface area that is to be considered.

The illustration shows the pressure control pipe connected to the steam main at the top as shown at *M*. This connection could be made at the bottom and the vertical dimension *F* reduced, in which case the reservoir would not be so important.

Experience has shown that a connection for this pressure control pipe at the bottom of the main steam line will allow sediment, pieces of packing and other foreign material to collect in the diaphragm chamber and connecting piping. In our opinion the proper place for the connection for the pressure control pipe is at the top of the pipe. Again, in some cases, the pressure control pipe is not connected near the regulator but is carried some distance away and sometimes it is carried upward. As the vertical dimension *F* is increased the application of a suitable reservoir is more important, if located at the highest point in the pressure control piping.

The diaphragm chamber and pipe connections can be drained as shown at *L*, where a valve can be placed, but as this is not a frequent operation, a plug is all that is needed.

The pressure gage is shown at *G*, where it gets the benefit of the water seal. For convenience it may be placed directly on the main pipe line, in which case the interior tube of the gage should be protected from the heat of the steam by a siphon coil. For very low steam pressures a gage of the compound type graduated for both pressure and vacuum is satisfactory.

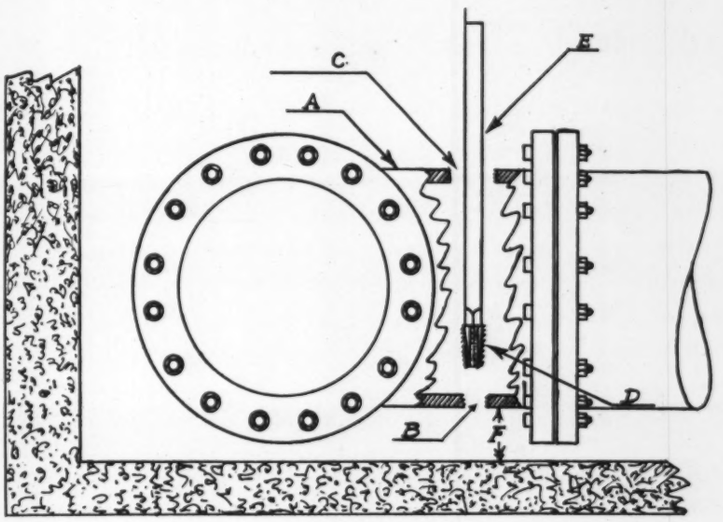
Box unions, ground joint pattern, should be placed as shown, so that in case it is necessary to work around the regulator, the pressure control piping can be easily disconnected and placed to one side out of the way.

The Relief Valve

Working even under the best of operating conditions, there are a few times when things go wrong. To insure safety in case the regulator should fail to shut off and permit the pressure to increase on the low pressure piping, a suitable and reliable relief valve should be installed as shown at *I*. The type of valve selected should have the outlet as shown so that in case the steam escaping from the valve would be liable to damage the building or its contents, the outlet could be piped to a place where it would not be detrimental. Precautions should be taken so that the outlet pipe will not freeze or become otherwise obstructed. A relief valve with a hand try-lever allows the valve to be periodically tested in order to determine if it is fast or corroded to the seat.

Tapping a Drain Opening

Pipefitters, working at maintenance work, are frequently called upon to work at jobs in very limited places. The accompanying sketch shows a method of tapping a drain opening on a large pipe line where there is very little room between the lower side of the steam main and the concrete floor of the tunnel in which the steam main is located.



Drain Opening Tapped Where Working Room Is Limited

In this particular case a large steam main was found to be low at one of the 90 deg. turns, shown at *A*, where the steam main was installed in a concrete tunnel, with very limited working space, other obstructions being present in the way of additional pipe lines.

The dimension of the space shown at *F* was just enough to put in a short pipe nipple and an elbow to provide for the drain valve, but not enough to permit drilling a hole for a 1¼-in. pipe tap and tapping the opening from the bottom. So, first a hole was drilled for a 2-in. pipe tap, on the top of the main shown at *C*, and this opening tapped for a pipe thread, which was later plugged when the job was finished. A hole was then drilled for a 1¼-in. pipe tap in the bottom of the elbow, or bottom of the main shown at *B*, by using a drill with an extension stem and working through the 2-in. pipe tap at the top of the main. The next move was to take a piece of 1¼-in. extra strong pipe, long enough to reach to the lower opening and, by a little handy forging work, one end of the pipe was formed to fit the end of the pipe tap and serve as an extension stem or wrench. This enabled the pipe fitter to start the tap in the lower opening by working through the top opening as shown at *E* and *D*.

The tap was run in the lower opening until it was two threads through on the lower side. It was then with-